EXHAUST EMISSIONS FROM A SINGLE-CYLINDER, SPARK-IGNITION ENGINE FUELED WITH GASOLINE, METHANOL, AND ETHANOL

A THESIS

Presented to

The Faculty of the Division of Graduate

Studies and Research

By

Samuel O. Lowry

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Mechanical Engineering

Georgia Institute of Technology

June, 1974

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Approved:

R. S. Devoto, Chairman

P. Durbetaki

Michael J. Matteson

Samuel V. Shelton

Date Approved by Chairman May 21, 1974

ACKNOWLEDGMENTS

I wish to express my sincere appreciation to Dr. R. S. Devoto for the knowledge and friendship he shared during the course of this project.

Thanks also go to Dr. P. Durbetaki, Dr. Samuel V. Shelton, and Dr. Michael Matteson for serving as members of the Reading Committee.

Finally, I wish to dedicate this thesis to my wife, Sheryl, whose love and understanding has enabled me to successfully complete this work.

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SUMMARY

Because of the undesirable health effects of automobile exhaust emissions, considerable attention has been focused on emission control systems. While the predominant method to date for reducing exhaust emissions is by engine modification, a possible alternative may be the use of substitute fuels. Alcohol fuels, in particular, appear to have certain distinct advantages. This study compares the exhaust emissions from an engine fueled with two alcohol fuels, methanol and ethanol, with the emissions from a reference blend of gasoline, Indolene. Emissions testing was done at two different speeds for each of the three fuels. Pollutants of specific interest are unburned hydrocarbons, carbon monoxide, and oxides of nitrogen. Fuel economy and power output with these fuels are also measured and compared.

Mass emissions of unburned hydrocarbons were comparable for Indolene and ethanol, while those with methanol were somewhat lower. Considerably more oxides of nitrogen were emitted by the engine when fueled with Indolene than with either of the alcohols. At 1000 rpm, the oxides of nitrogen with methanol peaked slightly higher than ethanol, while the reverse was true at 1800 rpm. Mass emissions of carbon monoxide were found to be about the same for all mixtures for all fuels. Fuel consumption of methanol was approximately twice that of Indolene, while that of ethanol was around 60% more than Indolene. Power output was approximately the same for all fuels.

CHAPTER I

INTRODUCTION

Automobiles and Air Pollution

An aroused social awareness of the ever-increasing ill effects of environmental pollution upon the public health has focused considerable attention on the standards of environmental quality under which we live today. This rapidly growing public concern over environmental control in the last ten years has resulted in the creation of the Environmental Protection Agency by the Federal Government, and the enactment of the Clean Air Amendments of 1970 by the Congress. Arresting the ever-increasing amounts of pollutants thrown into our atmosphere daily has been a dominant cause celebre in the public eye. The public furor may be attributed to such remarks as:

> The day may come soon--if it's not already here--in which the individual automobile can no longer be tolerated as a convenient form of transportation, simply because of its adverse effects on the health of people, not just the aesthetics of the atmosphere (1).

Indeed, the individual automobile with her internal combustion engine has been proved to be a primary contributor to atmospheric pollution, and without appropriate engine emission controls, will remain as such for the near future (2).

Because of its rather unique weather and atmospheric conditions, Southern California became the first geographical area to experience serious adverse effects from automobile exhaust emissions. A. J. Haagen-Smit established the fact that certain automotive exhaust

emissions underwent chemical changes in the presence of ultraviolet radiation to contribute to the infamous Los Angeles smog (3). California passed the first air pollution control legislation in the United States, and the precedent was soon followed by the Federal Government. Since the individual automobile has already been singled out as a primary polluter of the air, much time and effort has been spent in search of methods to clean up vehicular engines. The American automobile has so entrenched itself in the lifestyle of the average American family that it would be hard to imagine this nation properly functioning without the millions of vehicles that travel its roads daily. There remains, therefore, only one alternative if this country cannot tolerate automotive engines because of environmental and/or health considerations. The pollutants spewing forth from the internal combustion engine must necessarily be reduced to an amount conducive to the preservation of a healthful environment.

Uncontrolled vehicular internal combustion engines emit pollutants from three primary sources. Evaporation losses from the carburetor and fuel tank amount to approximately 18%, crankcase vents lose approximately 20%, and the remaining 62% is discharged in the exhaust. Fuel evaporation controls have been placed on all new cars made since 1971, and crankcase emittants have been routed back to the carburetor and burned on all new cars since 1964. The engine exhaust, therefore, remains the most significant contributor of pollutants from automobile internal combustion engines (4).

Many various types of pollutants are to be found in vehicular exhaust gases. Vehicular pollutants include unburned hydrocarbons,

carbon monoxide, carbon dioxide, oxides of nitrogen, particulate matter (smoke), lead, sundry odors, and oxides of sulfur (2). Those which contribute most seriously to atmospheric pollution are the unburned hydrocarbons (HC), carbon monoxide (CO), and the oxides of nitrogen (NO_x). Unburned hydrocarbons and oxides of nitrogen are responsible for the Los Angeles smog phenomenon. Carbon monoxide renders inactive a percentage of the hemoglobin in the blood, and prolonged exposure can be fatal.

The three principal pollutants from automotive engines pose a unique problem for scientists. Methods for reducing one of the pollutants may subsequently increase another, and vice versa. Thus, to a certain extent, emissions control becomes an acceptance of the lesser of two evils. For example, low levels of carbon monoxide are present in lean mixtures of air and fuel. However, maximum NO_x emissions are found at around 10% excess air. Conversely, rich mixtures of air and fuel produce smaller amounts of NO_x, and larger concentrations of CO. Catalysts to prevent formation of NO_x may accelerate the formation of carbon monoxide, or the carbon monoxide may break down the NO_x-reducing catalyst in a short period of time, thus rendering the catalyst unserviceable. Unburned hydrocarbons are primarily the result of wall quenching, and may be reduced to a large extent by better fuel distribution and better flame propagation within the cylinders (5, 6). Emission of unburned hydrocarbons can also be reduced by increasing the combustion chamber surface temperatures. However, increased combustion chamber surface temperatures will eventually lead to other engineering and design problems (7).

Federal Emissions Standards

Although they have since been revised, the Clean Air Amendments of 1970 first established the air quality standards the automotive industry was required to meet. Currently the 1975 model year standards are 1.5 gm/mile for hydrocarbons (0.9 gm/mile in California), 15.0 gm/mile for carbon monoxide (9.0 gm/mile in California), and 3.1 gm/mile for oxides of nitrogen (2.0 gm/mile in California). Standards for model year 1976 are lowered to 0.41 gm/mile for hydrocarbons, 3.4 gm/mile for carbon monoxide, and 2.0 gm/mile for oxides of nitrogen for the entire nation. Standards for model year 1977 are identical to those for model year 1976 except that oxides of nitrogen must be further reduced to 0.40 gm/mile nationwide (8). Table 1 depicts the percentage of reduction of automotive emissions required by the revised Clean Air Amendments compared to automobiles with uncontrolled emissions.

Table 1. Percent Reduction of Emissions Required by Federal Regulations from Uncontrolled Levels

Emission	1972 %	1973 %	1975* %	1975** %	1976 %	1977 %
HC	80	80	89	93	97	97
CO	69	69	82	89	96	96
NO_x	--	38	92	95	95	99

*Excluding California.
**California only.

The dramatic reduction in emissions required by the Clean Air Amendments has laid a great challenge before the automotive industry. In addition to the requirements for maximum levels of specific

pollutants, the amendments stipulate that the standards set forth must be met for five years or 50,000 miles by representative automobiles. Even if applied engine modifications bring emission levels below the Federal emission standards according to the established timetable, it will not be done without greatly increased costs to the public at large. Existing engine modifications have to date in most instances increased fuel consumption. The fuel shortages of the past year have been worsened, at least in part, due to the lower gasoline economy of engines laden with pollution control devices. As the automobile manufacturers continue to have great difficulty in meeting the requirements of the Clean Air Amendments, it will be imperative to examine all possible solutions in order to determine the one most practical.

Two major approaches have been considered by the automotive manufacturers in seeking to reduce automotive emissions. One approach is to abandon entirely the conventional gasoline engine in favor of a new power source (9). Candidates include steam engines (Rankine cycle), Stirling cycle engines, gas turbines (Brayton cycle), rotary engines, and engines powered by fuel cells or batteries. In general, these alternative power sources were found to require additional exhaust emissions control equipment to meet existing Federal air quality standards. Also in question is the capability of an assembly line-produced alternative power source to continue to meet or exceed the Federal air standards. The second approach, modification of existing engines, is receiving the major impetus. Engine modifications are the primary means of limiting exhaust emissions on 1973 and 1974 model year cars (10). Excellent surveys of the automotive exhaust emission

control techniques applied by the major automobile manufacturers are to be found in references (11), (12), (13), and (14).

More refined modifications for 1975 and 1976 model automobiles are still required in order to fully comply with the up-dated Clean Air Amendments, and extensive research continues in many aspects of automotive engineering. A variable-displacement engine has attracted some attention (15). This engine has been shown to be a feasible solution to the exhaust emission problem. Under varying driving conditions, the compression ratio will be automatically changed to provide for the maximum obtainable engine efficiency. A constant compression-ratio engine cannot be designed to operate at maximum efficiency under all types of driving conditions. Investigations have also been made of the effects of compression ratio, mixture strength, spark timing, and coolant temperature on exhaust emissions (16), of the effect of intake-air humidity, temperature and pressure on exhaust emissions (17), and of the effect of ambient laboratory conditions (18). These studies were either nonconclusive, or merely substantiated previous works or theory concerning exhaust emission formation. These papers failed to suggest any concrete methods to actually curtail automotive exhaust emissions.

Substitute Fuels

One solution to the exhaust emission problem which has received relatively scant attention has been the introduction of substitute fuels. The suggestion of alternative fuel sources for internal combustion engines is not really new, and may have been overlooked due to

the heavy emphasis on engine modification techniques for controlling automotive exhaust emissions. One reason for the preference of the "quick-fix" attitude to emission control is that not all authorities are convinced extensive emission control is needed (19). In any event, unless the Clean Air Amendments are completely revoked, the automobile industry must comply with Federal air quality standards, and the use of substitute fuels may be part of the answer to the problem.

Relatively cheap, natural gas has been the most prominent candidate for use as a substitute fuel. However, the most widespread use for natural gas seems to be in stationary power sources rather than automotive applications, primarily because of its limited availability. References (20) and (21) are two studies of the use of natural gas in internal combustion engines. Use of natural gas does provide a reduction of some emissions when compared to gasoline.

Studies of ammonia for use as a possible fuel have also been made. Ammonia was found to be capable of burning in an internal combustion engine if introduced in the vapor phase, and partially decomposed to hydrogen and nitrogen before ignition in the combustion chamber. Power output, however, was reduced from 70-77%, and specific fuel consumption was increased approximately two-fold (22). A separate study determined that ammonia and the combustion products of ammonia were compatible with existing automotive engineering materials and lubricants (23).

Alcohol fuels, in particular, have been used previously in internal combustion engines. The most serious attempts to use alcohol as a motor fuel in this country were in 1922 and 1923 when post-war

stocks of alcohol were high, and gasoline supplies had not caught up to the motorized boom that was sweeping the country. In 1933, farmers sought alternative uses for excess farm products, and alcohol fuels again received much attention. In Europe, a few countries used alcohols extensively as motor fuels. Their reasons were to maintain a motor fuel self-sufficiency, since they had no large natural petroleum reserves, and to keep alcohol production at its peak during peacetime. During periods of war, alcohol was in great demand for munitions production and medicinal purposes. Because of excessive costs, the absence of any definitive technical advantages, and plentiful gasoline supplies, use of alcohol as a motor fuel never became widespread in this country (24). The rigorous Federal exhaust emissions standards and the recent gasoline shortages may well combine to make essential a reevaluation of the practicality of alcohol use as a motor fuel.

Alcohol fuels present certain distinct advantages. Storage and transportation procedures for alcohols are already well-developed. Alcohols generally have a high octane number, and could conceivably be used to raise the octane rating of gasoline by mixing. Also, no major redesign of current automotive engines would be necessary for them to burn alcohol fuels. The primary disadvantages of alcohol fuels are probable higher costs than gasoline and a lower heating value (Table 2). Although some alcohols may be comparable with gasoline on a price per gallon basis, the reduced heat content makes the alcohol more expensive for identical power output (25).

The Vulcan-Cincinnati Company is developing a product called methyl-fuel, a combination of methanol and other alcohols, as a more

Table 2. Properties of Fuels

Item	Isooctane	Indolene	Methanol	Ethanol
Formula	C_8H_{18}	$CH_{1.86}$	CH_4O	C_2H_6O
Molecular wt	114.2	13.86	32.04	46.07
Carbon to Hydrogen wt ratio	5.30	6.45	3.0	4.0
Carbon, % by wt	84.1	86.6	37.5	52.1
Hydrogen, % by wt	15.9	13.4	12.6	13.1
Oxygen, % by wt	0.0	0.0	49.9	34.8
Boiling point, F at 1 atm	211	200-230	148.3	172
Freezing point, F at 1 atm	-161	--	-144	-170
Vapor pressure, psia at 100 F	1.72	--	4.55	2.25
Specific gravity, 60F/60F	.692	.741	.792	.8038
Liquid density, lb/gal at 60F and 1 atm	5.77	6.18	6.61	6.71
Surface tension, dynes/cm at 68F and 1 atm	18.77	--	22.61	23.04
Heat of vaporization, btu/lb at 77F and 1 atm	117	--	503	396
Heat of combustion, btu/lb at 77F	20556	20300	9770	12780
Stoichiometric AF	15.11	14.52	6.44	8.96

pollution-free fuel than conventional fuels now in use. In boiler combustion demonstrations, no unburned hydrocarbons were detected, and NO_x and CO emissions were lower than for natural gas. In automobile emission testing of their methyl-fuel, Vulcan-Cincinnati states that exhaust pollution levels are well below those of current gasoline. The absence of lead and sulfur also makes methyl-fuel compatible with the catalysts planned for use in 1975 and 1976 model automobiles (26).

Ebersole and Manning (27) have concluded an extensive study comparing the exhaust emissions of a single-cylinder engine fueled with isooctane and methanol (CH_4O). They found that emissions of unburned hydrocarbons were substantially lower for methanol when compared to isooctane, that carbon monoxide emissions for rich mixtures were not quite as high as for isooctane, and that NO_x emissions were in general also lower. Their study showed that engine output was essentially the same as for isooctane, but this was attained with a significantly greater fuel consumption of methanol. Reference (28) discusses the performance and exhaust emissions of an American Motors Gremlin operating on methanol. Their experimental results serve to substantiate the report submitted by Ebersole and Manning (27), and methanol is believed by these authors to be a definite possibility in meeting the Federal pollution requirements for 1975-1976.

While the possibility of using a pure alcohol in an internal combustion engine remains remote, a combination of alcohol and gasoline may be a positive step towards effective exhaust emissions control. Reference (29) suggests the use of strict exhaust emission controls only in highly urbanized areas where exhaust emissions pose

the most serious threat to health and environment. If a sufficient reduction in emissions could be achieved, a fuel blend of gasoline and alcohol might possibly be an attractive alternative to expensive emissions control equipment. The feasibility of this or related solutions will only be determined after extensive research and experimentation with alcohol fuels and blends.

Exhaust emissions are known to depend mainly upon compression ratio, air-fuel ratio, and spark timing. The single cylinder Cooperative Fuel Research (CFR) engine, discussed more fully in Chapter II, enables the rapid and convenient variation of these important parameters. For these reasons, the CFR engine was chosen as the standard engine on which to run the emissions tests of a reference gasoline and two representative alcohols.

The purpose of this research was to compare the exhaust emissions of two basic alcohols to those from a reference blend of gasoline. Since the time limitation in this work did not permit the variation of all parameters, much more work remains to be done with the same fuels. However, exhaust emissions for various air-fuel ratios and maximum power spark advance were obtained. Fuel economy and power output were also recorded and the data compared. The reader is cautioned that no attempt has been made to conduct an economic analysis of the impact of a widespread use of alcohol fuels on the economy. Hopefully this work, in conjunction with more detailed and extensive undertakings, will be a positive step toward reestablishment of a healthful, relatively pollution-free, environment for all to enjoy.

CHAPTER II

DESCRIPTION OF EQUIPMENT

Measurement and control of many parameters were essential to the collection of accurate and complete exhaust emissions data. The adoption of the CFR engine for exhaust emissions testing made several modifications of the original engine arrangement necessary. In addition to measuring the specific exhaust emittants, means for measuring or controlling oil temperature and pressure, air-intake temperature, changes in air-fuel ratio (approximate only), mixture temperature, coolant temperature, power output, and engine speed were essential. The scientific equipment and engine modifications required to be made for this work are described in detail in the remainder of this chapter.

Basic CFR Engine-Dynamometer Test Arrangement

The engine utilized in this work was a 1947 single-cylinder Cooperative Fuel Research (CFR) engine, normally used to determine octane ratings of various fuels according to reference (30). However, the CFR engine has frequently been used for exhaust gas analysis (16, 27, 31, 32, 33). The CFR engine permits the rapid and convenient adjustment of both the compression ratio and spark advance, an advantage not enjoyed by multi-cylinder engines. In addition, the CFR engine may be outfitted for fuel injection, and some work has been done with a combustion prechamber (34). This versatility of the single cylinder CFR engine makes it ideal for exhaust emissions testing. Pertinent

engine data is displayed in Table 3.

Table 3. Test Engine Data

Type	CFR engine, continuously variable compression, one-cylinder
Displacement	37.33 cu. in.
Compression ratio	4 to 10:1
Bore	3.25 in.
Stroke	4.50 in.
Weight of engine, approx.	650 lb.

Integral to the CFR engine before modification for exhaust emissions testing were intake-air and mixture rheostats for controlling intake-air and air-fuel mixture temperatures. Thermometers were placed on the engine to facilitate the recording of intake-air temperature, air-fuel mixture temperature, and coolant temperature. Engine oil pressure and engine oil temperature were read from gauges on the CFR control panel. An oil heater was used to achieve operating temperatures more quickly, and water lines were opened as necessary to bring oil temperatures back down into the desired operating range. Finally, maximum power spark advance was obtained by manual adjustment of the spark timing indicator located below the fuel bowls. The use and location of these components are fully described in reference (30).

Adjustment of the compression ratio setting was accomplished by manipulation of a micrometer and handcrank. A micrometer setting of

0.192 inches will produce a compression ratio of 7.5:1. This compression ratio setting remained constant for all the data collected in this thesis. Micrometer settings for other compression ratios may be found in reference (30).

A Meriam Laminar Flow Meter Model 50MC2 in conjunction with a Dwyer Inclined Manometer Model 102.5 was connected to the intake-air surge tank to measure the intake-air flow rates to the engine. The laminar flow element produces a differential pressure which varies linearly with air flow across a honeycomb matrix. Once the differential pressure is read from the inclined manometer, the air flow rate in standard cubic feet per minute (SCFM) may be found from the calibration curve accompanying the laminar flow element. Correction factors for temperature and pressure in the laminar flow element instruction manual were applied to air flow rates obtained at other than the reference conditions of 70 F and 29.92 in Hg.

Several modifications to the original engine were required before exhaust emissions testing could be started. In the original engine arrangement, the CFR engine was connected by a pair of belts to a synchronous induction motor. These belts were removed prior to any emissions testing since operation was desired at a higher speed than the induction motor allowed. A third belt connected the synchronous induction motor to a small direct current generator behind the CFR control panel. This generator provided the electrical current to the engine spark plug. Removal of the induction motor belts left the generator without a source of power, and the engine with no spark. An alternate means of providing current to the spark plug was made

possible by the installation of a Heathkit Regulated Power Supply Model IP-32. Future experiments may be conducted using current from either the Heathkit power source, or the induction motor and direct current generator, by the utilization of a permanent switch installed on the CFR control panel.

Vertical adjustment of the CFR engine fuel cylinders provided a coarse method for the adjustment of fuel flow into the engine. However, a more precise control of the fuel flow was needed. A Fischer and Porter Series 10A3135 Rotameter with needle valve was inserted into the fuel line, and fine adjustments were thus able to be made with the needle valve. Originally, it was hoped to read fuel flow rates directly from the rotameter, but given the range of flow rates and air-fuel ratios desired for the three fuels, sufficient accuracy was simply not obtainable. A smaller fuel bowl etched with graduations was used to estimate the amount of change of fuel flow, and hence, the change in air-fuel ratio, during the data-collecting process. A more detailed account of the data-taking procedure may be found in Chapter III.

In order to compensate for the different densities of the three fuels used in the emissions testing, new fuel jets were fabricated by the machine shop. Using the fuel jet size for gasoline within the CFR engine as a standard, calculations were made to determine the correct jet sizes for methanol and ethanol in order to produce the proper range of air-fuel ratios at both operating speeds. The jet sizes used for each fuel are shown in Table 4.

During the initial stages of data collection, the CFR engine exhaust manifold was discovered to contain several large holes, which

Table 4. Fuel Jet Sizes

Fuel	1000 rpm	1800 rpm
Indolene	.026 in dia	.026, .032 in dia
Methanol	.038 in dia	.037, .038 in dia
Ethanol	.035 in dia	.038, .035 in dia

lead to inaccuracies in the data that had been collected. A new exhaust manifold was fabricated to replace the leaking exhaust pipe, and all previous data was discarded.

The exhaust emissions sampling train (to be discussed more fully later) was connected to the exhaust pipe of the CFR engine by means of a 1/4-inch tap drilled into the pipe. This tap was threaded and an 8-inch stainless steel probe was inserted through the tap into the exhaust pipe. A Swagelok adaptor was welded to the probe and connected to a stainless steel line leading to the sample train system. The exhaust gases coming from the engine cylinder that did not enter the probe were carried out a previously existing exhaust pipe and discharged into the atmosphere outside of the engine room.

The power output of the CFR engine was absorbed directly by a General Electric Fuel Research Dynamometer Unit coupled directly to the CFR engine output shaft. Loading of the dynamometer was varied by adjustment of rheostats on a General Electric Dynamometer Control Panel. Loads were determined by use of a 0-50 pound scale adjacent to the dynamometer unit. The scale was graduated in 0.05 pound

increments. According to the data plate on the dynamometer, the equation for computing horsepower is given as:

$$\text{Horsepower} = \frac{(\text{lbs pull}) \times (\text{rpm})}{6000} \quad .$$

The dynamometer control panel, dynamometer, and scale were integral components of the original CFR testing apparatus, and as such, were securely bolted to the concrete floor.

Measurement of the engine speed was performed with a Hasler Speed Indicator to obtain a precise determination of engine revolutions per minute. This reading was taken directly from the dynamometer output shaft.

Emission Instrumentation and Sample Train

The appropriate exhaust emissions testing equipment was available in order to perform all necessary measurements prior to the start of the project. A major task was to disassemble, move, and reassemble the exhaust emissions apparatus in the present CFR engine laboratory. The exhaust emissions equipment had been used previously in the conduction of exhaust emissions testing on other engines and with other fuels. All instruments had been integrated into a single unified system called the sample train. The principle components of the sample train were nondispersive infrared (NDIR) analyzers for measuring nitric oxide, carbon dioxide, and carbon monoxide, a flame ionization detector (FID) for measuring total amounts of unburned hydrocarbons in the exhaust, and an oxygen analyzer for determining the oxygen content of the exhaust stream. The fundamental methods of operation and basic

calibration techniques for each of these instruments will now be discussed separately.

Hydrocarbons (FID)

A Beckman Model 400 Hydrocarbon Analyzer was used to determine the concentration of hydrocarbons present in the exhaust stream. The theory of operation of the hydrocarbon analyzer is one of flame ionization detection. The flame formed when hydrogen is burned in air contains a negligible amount of ions. When the exhaust sample is passed through the hydrogen flame, an ionization current is produced. This ionization current is proportional to the rate at which carbon atoms enter the burner, and, therefore, will provide the concentration of hydrocarbons in the original sample via direct read-out on a front panel meter. Accuracies of $\pm$2% of full scale are claimed by the manufacturer (35).

A mixed fuel of 41.5% hydrogen and 58.5% nitrogen was supplied to the burner at 29 psig. Use of the optional mixed fuel accessory provides for a better instrument response when there may be a variable oxygen concentration in the exhaust sample. This variable oxygen content is particularly evident when working with vehicular exhaust emissions.

The burner flame also required that compressed air be supplied at 20 psig. An optional automatic fuel shut-off served to arrest fuel flow in the event that the burner flame somehow became extinguished. Pressure regulators internal to the flame ionization detector were employed to regulate the mixed fuel, compressed air, and sample gases

passing through the instrument. The mixed fuel, compressed air, and calibration (span) gases were stored in high pressure cylinders equipped with high pressure regulators.

The procedure for calibration of the flame ionization detector is very elementary. A zero point is set by using the compressed air as an essentially hydrocarbon-free zero gas, and then a span point is set using a known mixture of propane (C_3H_8) in nitrogen. All span concentrations were multiplied by three to produce meter readings on a per carbon atom basis. In the literature, an alternative procedure is to furnish hydrocarbon data on a hexane (C_6H_{14}) basis. All hydrocarbon data in this thesis may be divided by six to obtain the hexane equivalent.

Since instrument response is linear throughout all ranges, once any span point is set, the sample concentration may be read directly from the front panel meter. Two span gases were used in calibrating the analyzer. The first span gas had a concentration of 9500 ppm propane in nitrogen (27500 ppm carbon), and was used until exhausted. The second span gas had a concentration of 1050 ppm propane in nitrogen (3150 ppm carbon), and was used until the data collection was completed. Both span gases were supplied by Matheson Gas Products, and were certified to be accurate to $\pm$1 ppm.

Since the hydrocarbon analyzer does not actually "see" on a one-to-one basis the unburned alcohol fuel molecules passing through the burner, the effective carbon numbers needed to be determined experimentally for both methanol and ethanol. The effective carbon number is defined as the instrument response caused by an atom of the

given type divided by the instrument response caused by an aliphatic carbon atom. The analyzer response to an aliphatic carbon atom, such as Indolene, is unity. The response differs for alcohol molecules due to the atom bonding within the alcohol fuel molecule. The effective carbon numbers for methanol and ethanol on this particular flame ionization detector were determined to be 0.75 and 0.92, respectively. FID meter readings were then divided by 0.75 and 0.92 for methanol and ethanol in order to obtain hydrocarbon emissions data on a comparable basis with Indolene. The values of 0.75 and 0.92 for methanol and ethanol compared rather poorly to the approximate effective carbon numbers of 0.4 for methanol and 0.7 for ethanol given in the instruction manual. However, the value for methanol compares favorably to the effective carbon number of 0.85 found by Ebersole and Manning in reference (27). Appendix A describes in more detail the experimental procedure by which the effective carbon numbers were obtained.

Nitric Oxide, Carbon Dioxide, and Carbon Monoxide (NDIR)

Concentrations of nitric oxide (NO), carbon dioxide (CO_2), and carbon monoxide (CO) in the exhaust stream were measured by Olson-Horiba Model AIA-2 Non-dispersive Infrared (NDIR) Analyzers. A separate analyzer was used for each substance, and all three analyzers operated on a principle of infrared absorption. Infrared light from two identical sources passes through a rotating, slotted disc called the chopper, and passes into either a sample cell or reference cell. A portion of the infrared radiation is absorbed in the sample cell, and measured by a detector located at the end of the cells. The amount

of radiation absorbed is proportional to the volume concentration of the particular emission of interest. The detector causes a meter reading, and this meter reading is applied to the appropriate calibration chart for the instrument. The concentration in volume per cent (CO_2 and CO) or parts per million (ppm) (NO) may be obtained directly from the calibration curve.

Zero and upscale span points were preset in a manner similar to the flame ionization detector in order to properly calibrate the instruments. Compressed air was used as the zero gas for all NDIR analyzers. A mixture of 1.06% carbon monoxide in nitrogen was used as the span gas for the CO analyzer, and a mixture of 650 ppm nitric oxide in nitrogen was used as the span gas for the NO analyzer. Both span gases were supplied by Matheson Gas Products, and certified to be correct to $\pm$1 ppm.

A new calibration curve had to be drawn for the carbon dioxide analyzer in order that volume concentrations of approximately 15.0% carbon dioxide could be most accurately determined. A new calibration curve was drawn using three known span points according to the directions in the analyzer instruction manual. After the new curve was established, routine calibration of the CO_2 analyzer was performed with the 16.38% carbon dioxide in nitrogen span gas. Matheson Gas Products also supplied this span gas, and certified the gas to be correct to $\pm$1 ppm.

Additional instrument specifications and more detailed calibration and emissions measurement procedures for each of these three analyzers may be found in reference (36). Olson-Horiba states the

accuracy of these instruments to be ±1% of full scale and the sensitivity to be 0.5% of full scale.

Oxygen Analyzer

The Beckman Model 741 Oxygen Analyzer was used to determine the quantities of oxygen present in the exhaust gases. The oxygen content in the exhaust stream is commensurate with the response of an amperometric oxygen sensor, which responds to the partial pressure of oxygen. An amplifier unit measures the magnitude of the sensor signal, and the concentration of oxygen is read directly on the meter.

For the oxygen analyzer, there is no requirement to set a zero point as part of the routine calibration procedure. The zero point may be checked, however, by running any oxygen-free gas through the analyzer and checking to see that the meter registers zero. An upscale span point was set by assuming that the oxygen content of compressed air is 20.95%. The manufacturer states the accuracy of this instrument to be ±1% of full scale (37).

In spite of the fact that this instrument had recently returned from the manufacturer, the majority of data from this instrument is believed to be inaccurate. Very simply, consistent readings from this instrument were not obtainable. For example, there should be small quantities of oxygen in the exhaust (on the order of 0.0% to 0.5%) for fuel-rich mixtures of Indolene. For one run the meter would show the oxygen content in this range; for the next run, the oxygen content would be approximately 5.0% according to the analyzer. The same inconsistency was found to occur for all three fuels. The meter would zero by passing an oxygen-free gas through the sensor, so this

eliminated the possibility of a leak. The sensor head was also recharged, but to no avail. Previous users had also reported inaccurate data from this instrument.

Sample Train System

All of the exhaust-measuring instruments were consolidated into a unified system called the sample train. The exhaust gases entered the train directly from the tailpipe of the CFR engine. From this point, the sample train distributed the exhaust gases to each individual instrument in order that all required emissions measurements could be taken. The instruments themselves were located in a single cabinet housing the required valves, rotameters, and gauges to zero, calibrate, and record sample data in the most convenient manner. A schematic of the sample train system is shown in Figure 1.

From the tap drilled into the engine exhaust pipe, the exhaust gases were pulled through a condensing ice bath and water trap before entry into any of the analyzers. A Model MB 110-10 welded bellows vacuum pump/compressor, manufactured by the Metal Bellows Corporation, was used to pull the exhaust gases from the tail pipe through the ice bath and water trap. The ice bath consisted of a coil of 3/8-inch stainless steel tubing placed in a large stainless steel bucket. A mixture of ice and water was placed in the bucket to cool the exhaust sample and condense out most of the water vapor. The water trap downstream from the ice bath collected the condensed water vapor. The dry exhaust sample then passed through the pump and was distributed to each of the different analyzers.

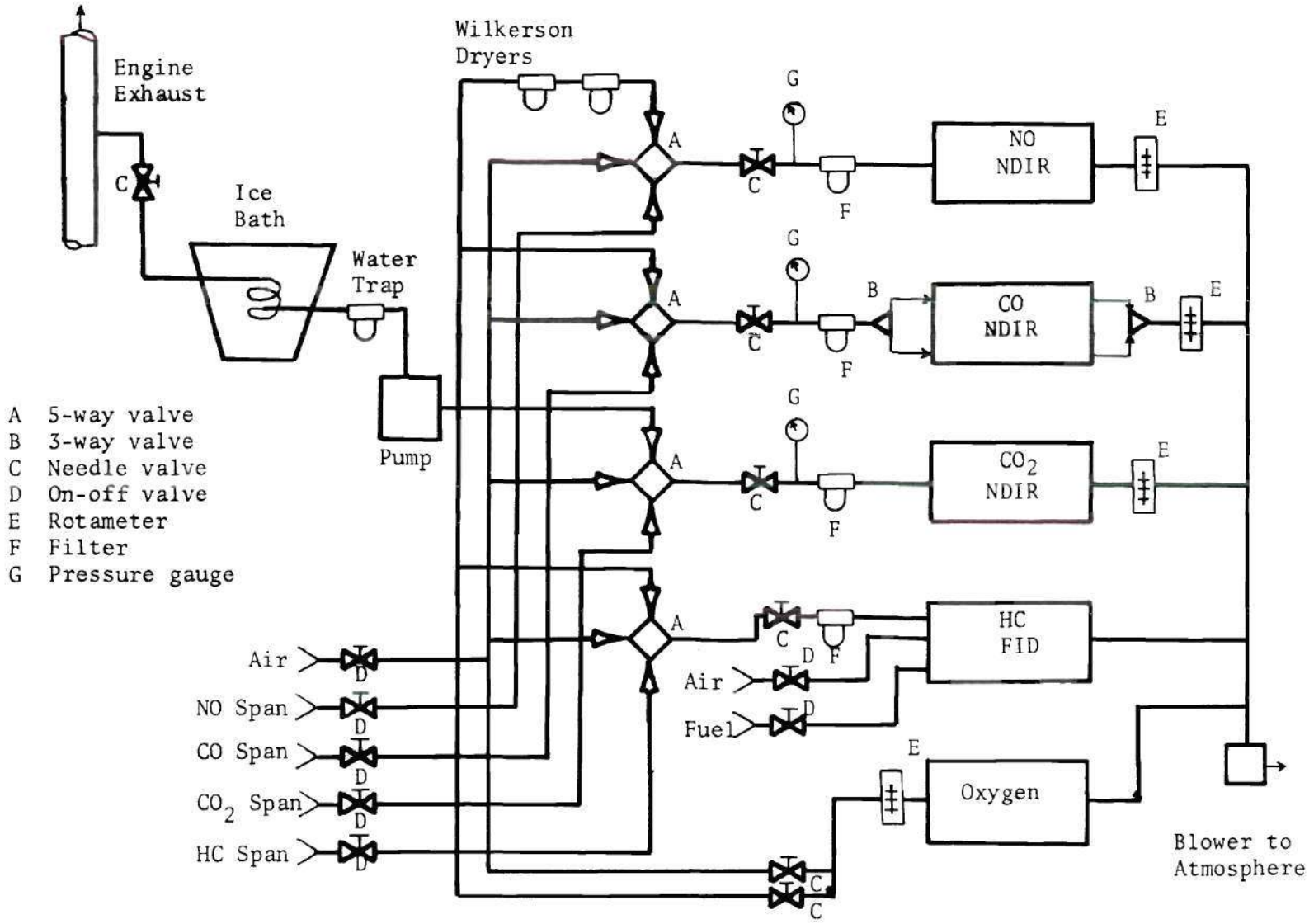

Figure 1. Sample Train

The portion of the exhaust sample destined for the hydrocarbon analyzer passed through one gate of a Whitey five-way valve and a Wilkerson Model 1049 particulate filter before entering the flame ionization detector. The other gates of the five-way valve were used for passage of span and zero gases to the FID. Flow rates of the zero and span gases, and the exhaust sample, were regulated by internal pressure gauges. Upon leaving the hydrocarbon analyzer, the sample gas passed through a squirrel cage blower to the atmosphere.

The portion of the exhaust sample directed toward the NO analyzer first passed through two Wilkerson Model 4001-2 dryers containing an indicating silica gel. These dryers were required only for the NO unit, since this analyzer was most sensitive to excessive water vapor. The silica gel had to be dried periodically in a vacuum oven to restore its water-absorbing capability. From the dryers, the exhaust sample passed through a Whitey five-way valve, a needle valve, a Wilkerson Model 1049 particulate filter, and then into the analyzer. Similarly to the hydrocarbon analyzer, the remaining gates of the Whitey five-way valve permitted entry of zero and span gases for calibration purposes. The needle valve was provided for sample throttling, and a Magnehelic 0-4-inch W.G. manometer measured inlet sample pressure. A Brooks 2-18 SCFH rotameter measured the exhaust sample flow rate upon leaving the NO analyzer, and before passage through the blower to the atmosphere.

The arrangement of the CO and CO_2 analyzers was almost identical to that of the NO analyzer. For the CO and CO_2 analyzers, the silica gel dryers were not required to be present. The Whitey five-way valves served the same purpose as for the NO analyzer, and both the CO and

CO_2 analyzer systems had needle valves to regulate the flow rates of the exhaust sample. A Magnehelic 0-8-inch W.G. manometer measured inlet sample pressure for the CO_2 analyzer, while a Magnehelic 0-4-inch W.G. manometer measured inlet sample pressure for the CO analyzer. Both systems also had Brooks 2-18 SCFH rotameters for determining flow rates of sample gases. Finally, the CO analyzer had three-way valves directly before and after the analyzer unit to allow for selection of two available sample cells. However, during the course of data collection, the settings on these valves were never changed.

Sample flow to the oxygen analyzer was determined by the use of two needle valves. One valve, when opened, allowed the exhaust sample to enter the oxygen analyzer. The second valve was necessarily closed. When the procedure was reversed, opening of the second valve allowed compressed air to enter the analyzer, and the upscale span point was set. As mentioned previously, there is no requirement for a zero gas with the oxygen analyzer. Between the two needle valves and the analyzer unit was a Brcoks 2-18 SCFH rotameter used to measure the sample flow rate into the analyzer. The sample gases exited from the oxygen analyzer, entered the blower, and were discharged into the atmosphere.

All zero and span gases had to pass first through Whitey off-on ball valves before entry into the five-way valves shown in the schematic. All valves were made of stainless steel, equipped with Swagelok connections. In general, instrument manufacturers have recommended the use of stainless steel or Teflon tubing in the sample train for exhaust testing. This recommendation was followed as closely as

possible, with copper tubing being utilized in certain instances. All tubing connections were made using Swagelok fittings to ensure leak-free connections and to withstand continual assembly and disassembly during this and future experiments.

CHAPTER III

EXPERIMENTAL TEST PROCEDURES

Once the engine was in satisfactory operating condition and the sample train connected, the data-taking procedure was begun. Exhaust emissions data was desired for three fuels at speeds of 1000 and 1800 revolutions per minute over a representative range of air-fuel ratios.

Fuels

The fuels used in this research project included Indolene 30, a standard reference fuel marketed by Amoco, with a carbon to hydrogen ratio of 1.86. The second fuel was Methanol (CH_4O), purchased from the Fisher Scientific Company. The third fuel was Ethanol (C_2H_6O), and this fuel was purchased from Burris Chemical Company. The trade name of this last fuel was Synasol (see Table 2). One 55-gallon drum of each fuel was purchased, and these drums were stored in a separate room away from the CFR engine itself. As needed, a one-gallon container was filled with the fuel being tested. The fuel cylinder of the CFR engine was filled with fuel from this one-gallon container, and during the course of data taking, was refilled quite often.

Warm-Up

A considerable amount of time was consumed prior to each day of engine operation to allow for equipment stabilization and engine warm-up. The exhaust emission instruments required one to two hours to warm-up in order to achieve stabilization. The NDIR instruments could

be left on overnight, and a warm-up period was avoided for them. However, in order to conserve the burner fuel supply, the hydrocarbon analyzer was shut down overnight and required one to two hours to achieve stabilization. The oxygen meter only required 5 to 30 minutes for warm-up.

The CFR engine also had to be warmed up properly before data taking could be started. Temperatures of the oil, intake-air, air-fuel mixture, and coolant had to be in the desired ranges.

In general, all exhaust emissions instruments were left on at least one hour. The engine was then started, and allowed to reach the desired operational temperature while the exhaust emissions instruments were calibrated. Once this was completed, the engine was close to or at the desired range for operation. The sample train was connected, ice and water placed into the ice bucket, and collection of emission data was started. No emission data was collected until the engine was properly warmed-up, and all instruments had been on for at least two hours.

Desired Operating Ranges

A survey of the literature of other CFR engines used for exhaust emission testing led to an arbitrary selection of desired operating ranges for intake-air temperature, air-fuel mixture temperature, and oil temperature. The intake-air temperature was adjusted to the range of 120 $\pm$ 5 F, and remained fairly constant for all fuels. Air-fuel mixture temperatures were more difficult to hold steady, and a range of 130 $\pm$ 15 F was selected. Because of the larger heat of

vaporization of methanol (Table 2), considerably more heat had to be supplied to the mixture manifold than for either of the other fuels. Since extensive use of the mixture heater was necessary, the mixture temperature for methanol was considerably more difficult to hold constant over the range of air-fuel ratios. The oil temperature was found to remain relatively constant for all fuels at the 1000 rpm level, but would increase much too rapidly when the engine was operated at 1800 rpm. Data runs at 1000 rpm and 1800 rpm were alternated to enable the engine to cool between 1800 rpm runs. This procedure and the selective use of the oil heater and oil coolant system kept oil temperatures easily with the 135 $\pm$ 5 F range. Methanol was found to be an exception, and numerous 1800 rpm data runs could be taken successively with no excessive oil temperature rise. Table 5 contains a summary of the engine operating specifications.

Table 5. Operating Specifications

RPM	1000 $\pm$ 25 1800 $\pm$ 60
Oil pressure	42 $\pm$ 4 F
Oil temperature	135 $\pm$ 5 F
Intake-air temperature	120 $\pm$ 5 F
Mixture temperature	130 $\pm$ 15 F
Coolant temperature	210 $\pm$ 2 F

Motor Test

Friction horsepower (fhp) was the first item to be determined, although not without some minor difficulty. Once the engine was in the desired operating range, as discussed above, motor tests were conducted to experimentally determine the friction horsepower. In the first weeks of data collection, several motor tests were conducted. The oil temperature, scale reading, and rpm were recorded and a plot made of fhp versus rpm (Figure 2). The fhp at both 1000 rpm and 1800 rpm were desired. The engine could not be motored below approximately 1200 rpm since the motoring generator was rated too low for the dynamometer and control panel. Motoring under this speed would have placed an excessive load on the generator set. The points in Figure 2 were extrapolated to 1000 rpm, as shown, and an average value of 1.5 for the friction horsepower was determined. At 1800 rpm, the fhp was determined directly from the figure to be 4.3.

Emissions Testing

As the first step in collecting emissions data, the engine was operated with the proper fuel jet at the desired speed with the fuel at the maximum level in the fuel bowl. All engine operation was done at wide open throttle (WOT). The engine speed was set by using a tachometer on the CFR control panel. The scale was balanced, and the spark advance adjusted to provide for maximum power. The engine speed was readjusted if necessary, and then data were recorded. The data, in order recorded, were differential pressure across the laminar flow element, oil pressure, oil temperature, air-intake temperature, fuel

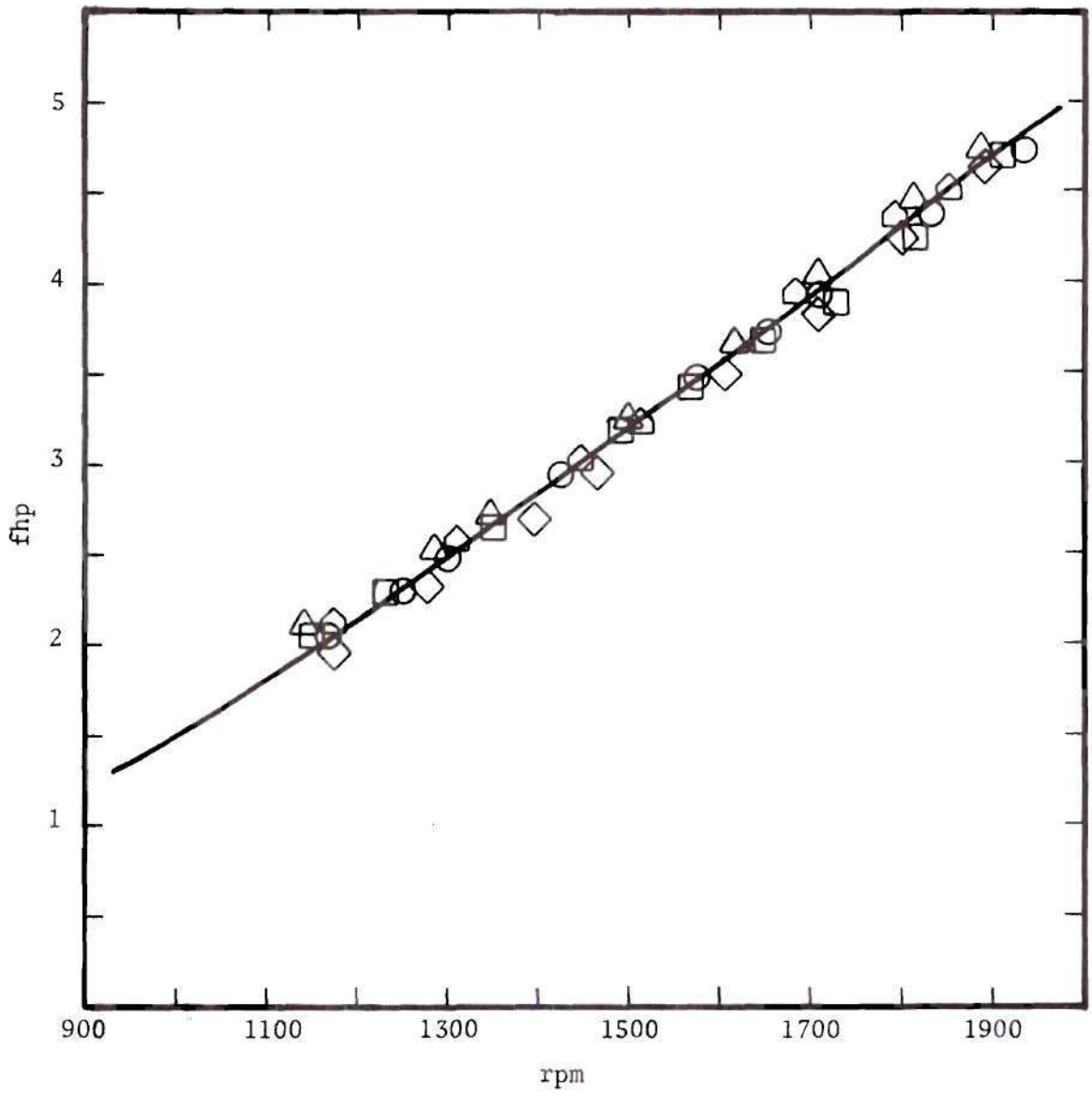

Figure 2. Motoring Friction Horsepower vs. Engine Speed

bowl setting, and spark advance. The fuel bowl setting was a relative scale setting, and was used to estimate changes of air fuel ratio only. Next, recordings were made of air-fuel mixture temperature, water coolant temperature, and scale readings of the load on the dynamometer. Generally, the scale needed to be rebalanced slightly. Once the scale was set, the flow rates of the exhaust sample were checked and readings were made of the meter readings from the FID hydrocarbon analyzer, NDIR instruments, and the oxygen analyzer. The actual engine speed was then determined by an average of two Hasler Speed Indicator readings taken from the dynamometer output shaft. Finally, the room temperature was recorded.

For the next run, the fuel flow rate was lowered by turning the needle valve on the Fischer rotameter in the fuel line. This caused a drop of the fuel level in the fuel bowl, and the graduated bowl provided an estimate of the amount of change in the fuel flow. Since the air flow rate remained relatively constant, this changed the air-fuel ratio to make the mixture slightly leaner. The entire process was repeated, starting with the check for maximum power spark advance (MPSA).

Five to eight data points were recorded before the fuel level fell below the bottom of the fuel bowl. These five to eight data points comprised one trial run. Test runs were repeated many times for each fuel and each engine speed to provide representative exhaust emissions charts for each fuel. No attempt was made to repeat identical air-fuel ratios on successive trial runs. A sample data sheet is shown as Appendix B.

NDIR meter readings and the respective calibration curves gave

volume per cent concentrations for carbon dioxide, carbon monoxide, and oxides of nitrogen. The oxygen analyzer meter read directly in per cent oxygen. Hydrocarbon analyzer meter readings were in ppm, and this was easily converted to a volume per cent concentration. FID meter readings for methanol and ethanol were divided by 0.75 and 0.92, respectively, to obtain hydrocarbon emissions on ppmC basis.

The dynamometer scale reading in pounds and the average Hasler Speed Indicator reading in rpm were converted to a value for brake horsepower (bhp). Adding the values for brake horsepower and the experimentally determined friction horsepower (fhp) gave the indicated horsepower (ihp) of the CFR engine.

At this point exhaust emissions in dry volume per cent concentration and brake and indicated horsepower data are available. But these data must be examined with regard to the air-fuel ratio. Two methods were used to determine the air-fuel ratio, and the results of these two methods are discussed in the next chapter.

The first method of calculating air-fuel ratios was by using the D'Alleva Charts (38, 39). This method was believed to be the most accurate for reasons explained later. The charts in reference (39) plot exhaust emissions versus air-fuel ratios for various ratios of hydrogen and carbon atoms in the fuel molecule. Using the same procedure as D'Alleva, similar charts were developed for Indolene ($CH_{1.86}$) and the two alcohol fuels. The derivation of these charts for a general fuel molecule of the form CH_yO_z is shown in Appendix C, and the charts derived from the D'Alleva method for Indolene, methanol, and ethanol are shown as Figures 30, 31, and 32, respectively.

According to the D'Alleva method, the exhaust emission of a particular pollutant was selected, and the dry volume per cent concentration of that pollutant determined the air-fuel ratio at which the engine was operating. Carbon dioxide was chosen as the particular pollutant, since its values for both rich and lean air-fuel mixtures provided the most convenient means to obtain the air-fuel ratio. Thus, knowing the dry volume per cent concentration of carbon dioxide for every data point led to the determination of the air-fuel ratio at every data point.

Carbon monoxide could also be used as the reference emission to determine the air-fuel ratio from the D'Alleva charts. For high equivalence ratios, use of carbon monoxide as the reference emission generally yielded air-fuel ratios which were slightly more fuel-rich than when carbon dioxide was used as the reference emission. Consequently, use of carbon monoxide yielded a smaller air-fuel ratio, and a larger fuel flow rate. The result is that emissions on a g/ihp-h basis are about the same for either of the two methods. The difference in mass emissions can be shown to be negligible by selecting an example where the difference in air-fuel ratios is relatively large. For one particular Indolene data run, use of carbon dioxide as the reference emission gives an air-fuel ratio of 12.6 and a fuel flow rate of 15.36 g/min. Mass emissions are found to be: HC, 2.57 g/ihp-h; NO_x, 1.92 g/ihp-h; and CO, 189.5 g/ihp-h. Use of carbon monoxide as the reference emission gives an air-fuel ratio of 11.8 and a fuel flow rate of 16.40 g/min. Mass emissions are found to be: HC, 2.59 g/ihp-h; NO_x, 1.94 g/ihp-h; and CO, 191.9 g/ihp-h. Therefore,

for the purposes of this work, the difference in mass emissions may be neglected, and carbon dioxide was used as the reference emission throughout.

A second and more exact procedure for the determination of air-fuel ratios is according to Spindt (40). Appendix D derives general formulas for the air-fuel ratios of alcohol fuels using the Spindt method which included the percentage of oxygen for lean mixtures. The continual erratic behavior of the oxygen analyzer led to the use of the D'Alleva method rather than that of Spindt for determining air-fuel ratios. A portion of the data obtained with the oxygen analyzer is felt to be accurate, and this data compared well to the D'Alleva method for determining air-fuel ratios.

Regardless of which of the two procedures described above were followed, once the air-fuel ratio was known, the fuel flow rate was very simple to find. The calibration curve for the laminar flow meter provided the air flow rate in SCFM and this was easily converted to grams per minute (g/min). The air flow rate and air-fuel ratio for each data point permitted the calculation of the fuel flow rate in g/min for each data point. The indicated specific fuel consumption (isfc) was calculated for each data point using the fuel flow rate ($\dot{w}_f$) and the indicated horsepower (ihp).

This procedure was duplicated for each data point, and enough data runs were conducted for each fuel and at each rpm to enable construction of exhaust emission charts.

Sources of Errors

Many sources of errors can be found in an experimental investigation such as this. The largest errors in this work probably resulted from malfunctions and irregularities in the exhaust emission measuring equipment. Although the equipment is certainly of adequate quality for this work, some problems were associated with every piece of equipment. The capillary tube became blocked on the FID, reducing HC emissions to zero. The response time of the NO NDIR is believed to be much slower than that claimed by the manufacturer. The CO NDIR was difficult to calibrate at times, and the oxygen analyzer never did operate satisfactorily. These major discrepancies were noted and taken into consideration. For example, data that was clearly inaccurate was completely discarded, and no oxygen analyzer readings are reported in this work. Therefore, these types of irregularities were not difficult to avoid. However, the extent to which minor irregularities may have influenced the data is not exactly known.

Other sources of error may have resulted because of the age and condition of the CFR engine. No attempts were made to overhaul or internally clean the engine before its use. Because of the age of the engine, the compression ratio may have varied slightly from 7.5. At 1800 rpm, engine vibration made adjustment for maximum power spark advance difficult. Precise scale readings were also extremely more difficult to obtain at this higher engine speed. Although some drift in engine speed was present, the Hasler Speed Indicator gave rpm readings to a sufficient degree of accuracy so that data runs at speeds outside the indicated operating tolerances were eliminated, and the data run

repeated. Almost no drift in engine speed was noticeable at 1000 rpm.

Some contamination may have been present in the sample lines, but any errors of this sort are felt to be minor. The sample lines were flushed with compressed air both before and after use. Contamination was much more likely to be found, however, in the stainless steel tubing leading from the exhaust pipe of the CFR engine into the ice bucket. Hydrocarbon deposits built up in the tubing over a period of time, and could have possibly affected some engine exhaust emissions.

Finally, the possibility of human error is present in all phases of the experimental procedure. This error is also felt to be small, since all instruments are calibrated to a sufficient degree of accuracy to allow for convenient reading of all data. Gross errors in meter readings would have been evident as computations were made for various data runs. Small errors in meter readings would not alter the conclusions of this project. Similarly, errors introduced because of the effects of ambient laboratory conditions, air humidity, human error in reading calibration curves, etc., will also not be of enough significance to alter the results presented in Chapter IV.

CHAPTER IV

RESULTS

The data used to construct the appropriate exhaust emissions charts in this chapter is shown in table form in Appendix E. All exhaust emissions have been converted to g/ihp-h using the procedure established by Stivender (41). An example of the conversion of emission data in dry volume concentration to a mass basis is shown in Appendix F. Mass emissions have been plotted against equivalence ratios for each fuel and at each speed. Average curves have been drawn through the points to indicate the approximate mass emissions levels for various equivalence ratios. Even in the most elaborate emissions laboratory, a wide scatter of emissions data can result, with variations of up to 50% due to changes in instrumentation, emission control systems, and variations in test procedures (42). However, valid comparisons of the exhaust emissions from the CFR engine operating on these fuels can be made with the representative emission curves shown in this chapter. Power and fuel economy with these fuels were also determined and are readily compared in chart form.

Emissions of HC

Mass emissions of unburned hydrocarbons from the engine fueled with Indolene, methanol, and ethanol are shown in Figures 3-8. Emissions of unburned hydrocarbons with methanol are less than for either Indolene or ethanol over the entire range of equivalence ratios.

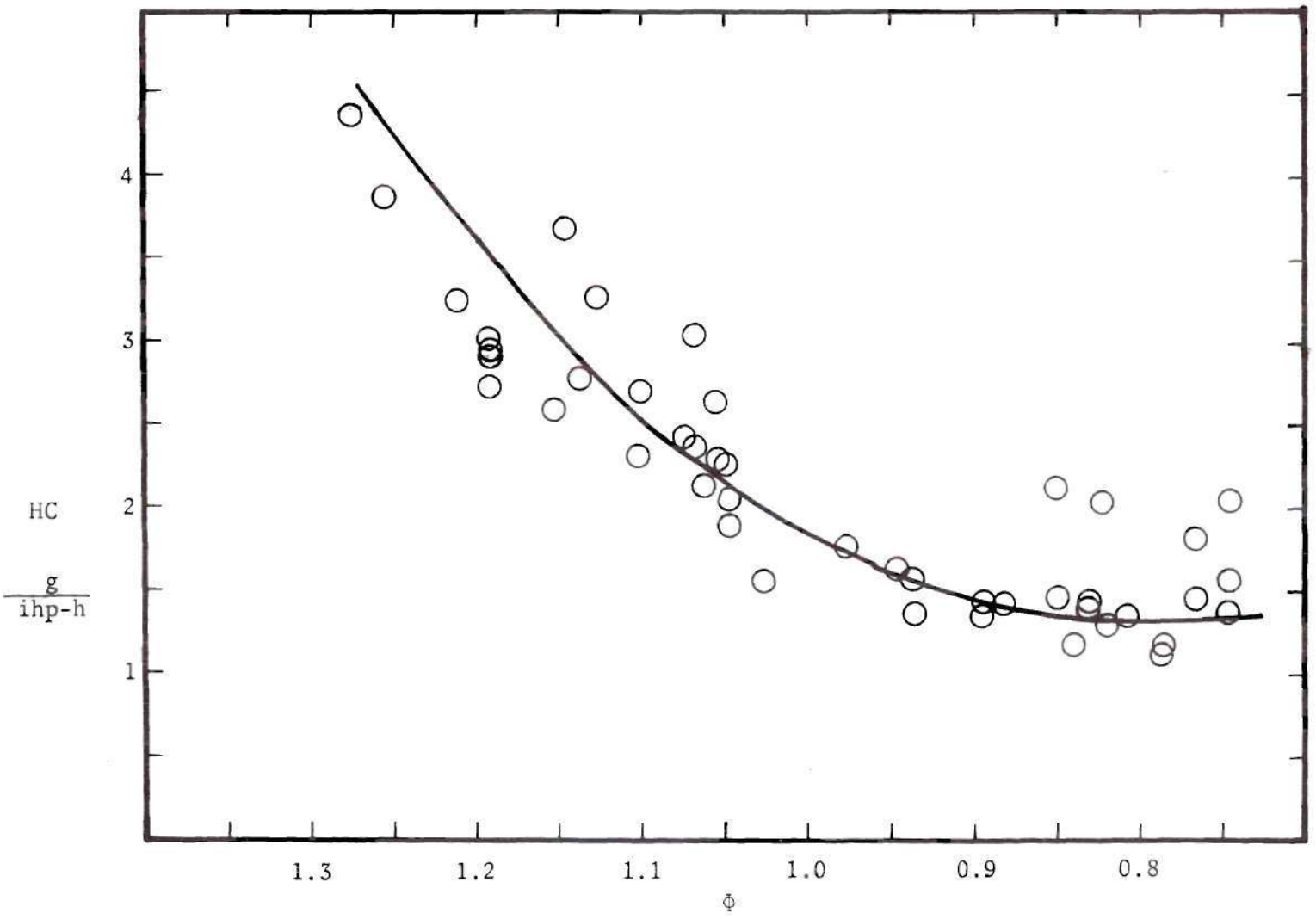

Figure 3. Indolene HC Emissions at 1000 rpm vs. Equivalence Ratio

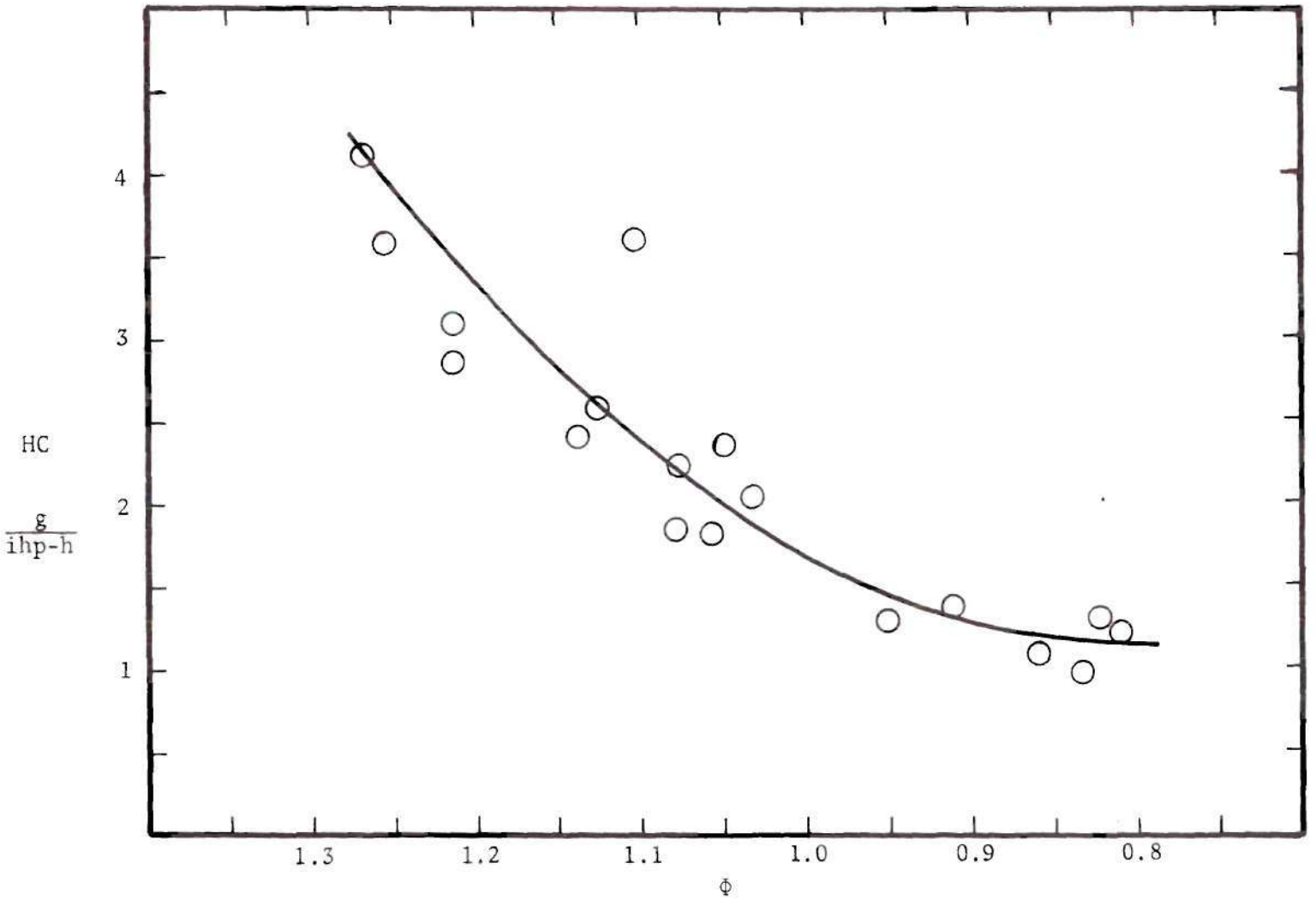

Figure 4. Indolene HC Emissions at 1800 rpm vs. Equivalence Ratio

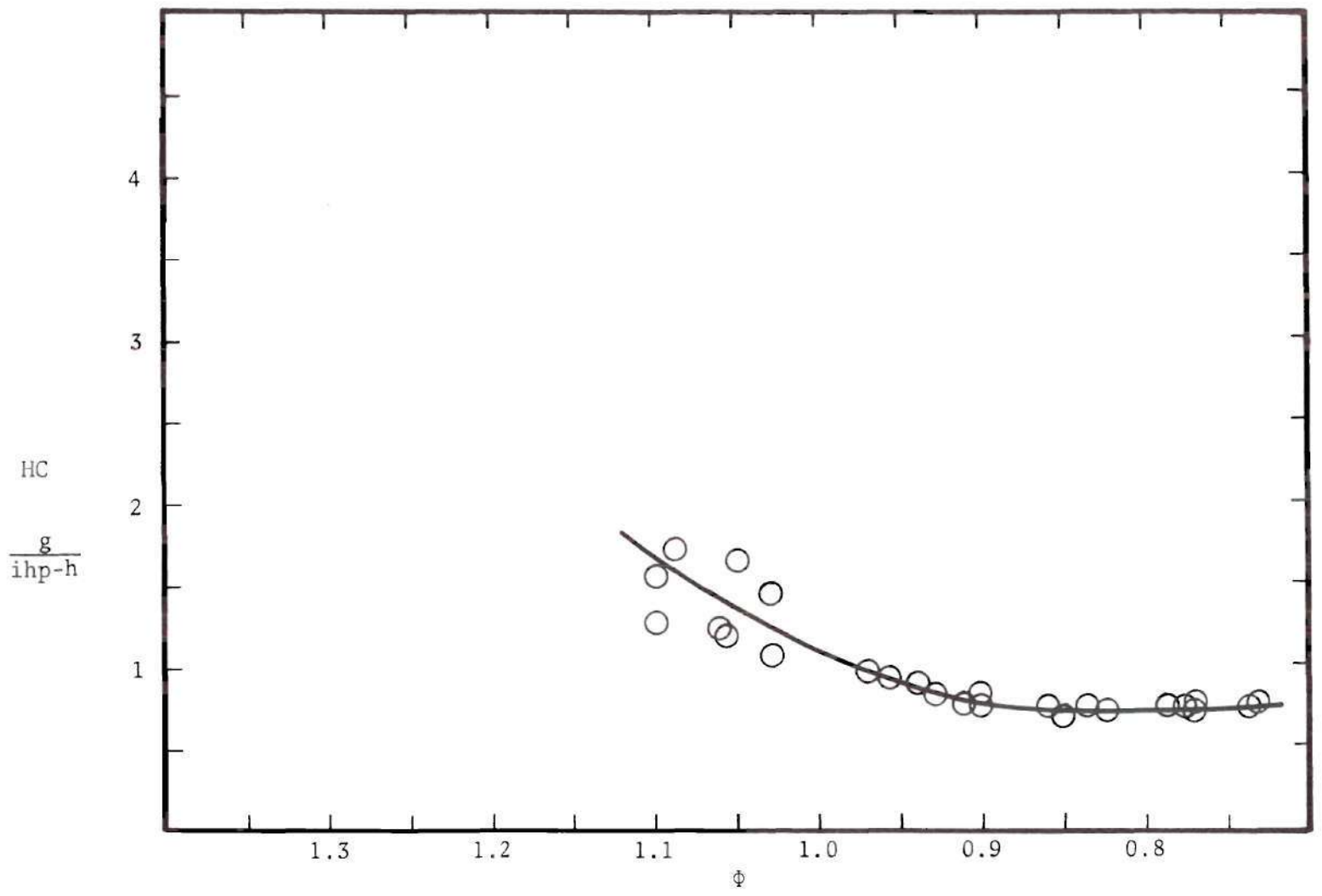

Figure 5. Methanol HC Emissions at 1000 rpm vs. Equivalence Ratio

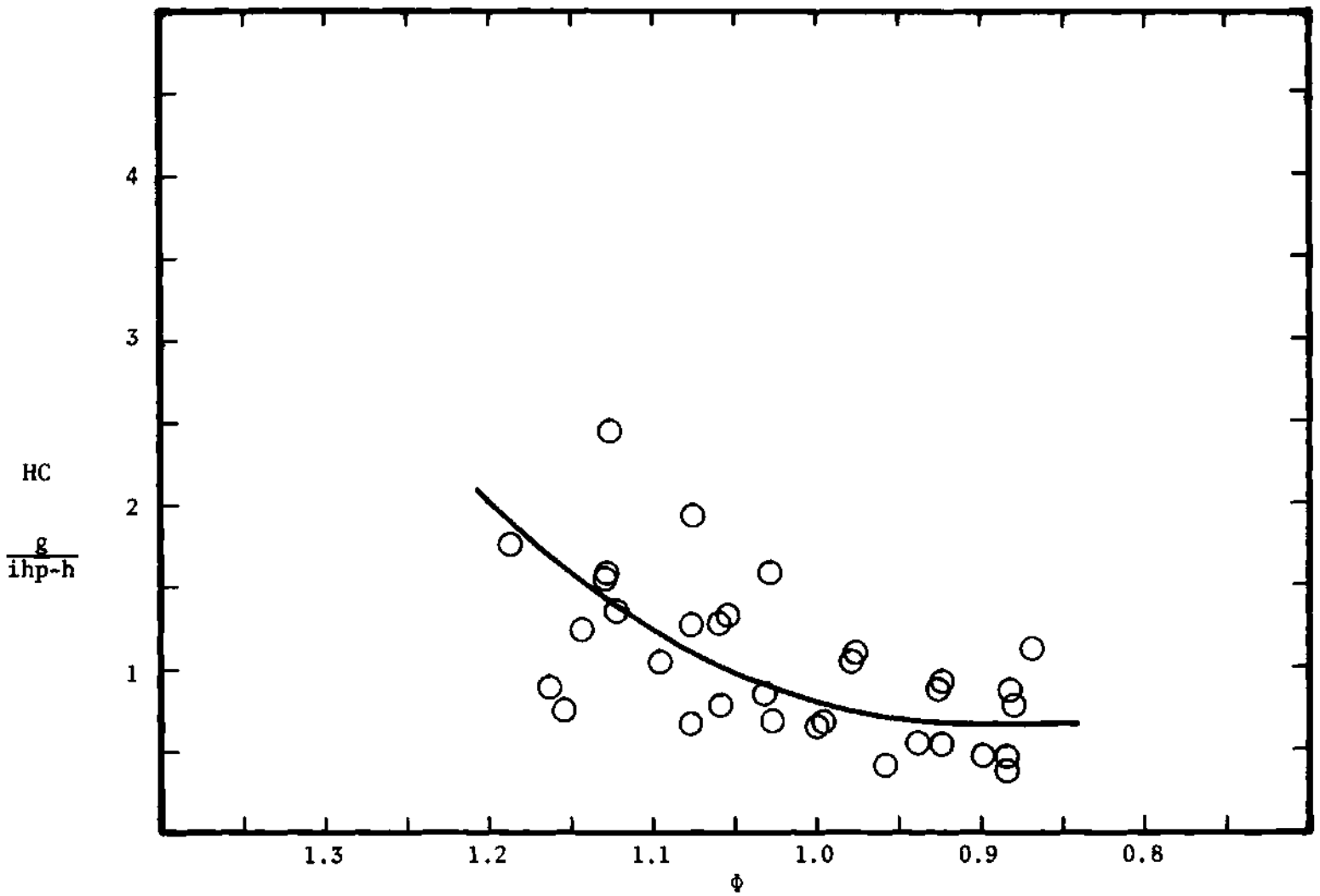

Figure 6. Methanol HC Emissions at 1800 rpm vs. Equivalence Ratio

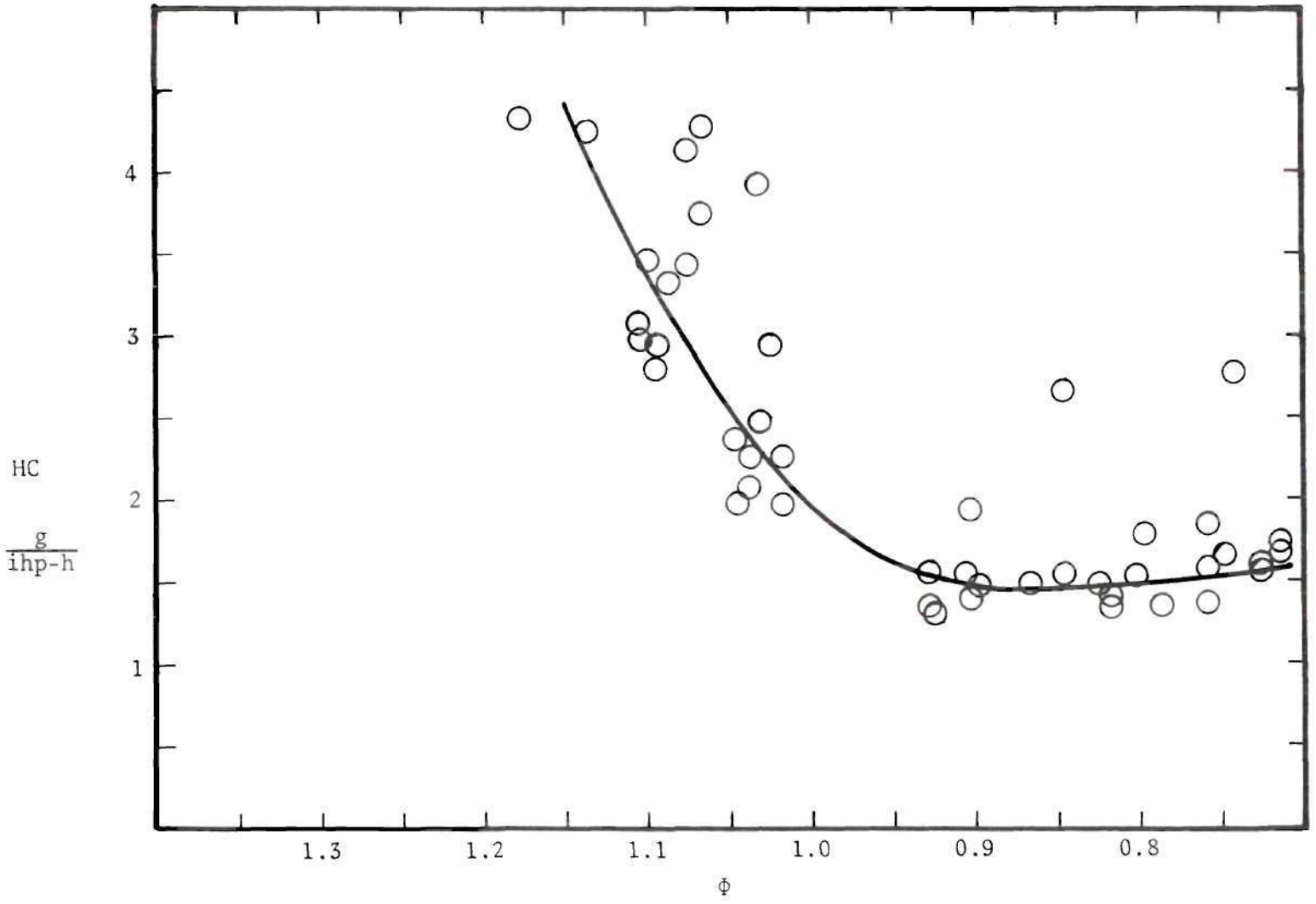

Figure 7. Ethanol HC Emissions at 1000 rpm vs. Equivalence Ratio

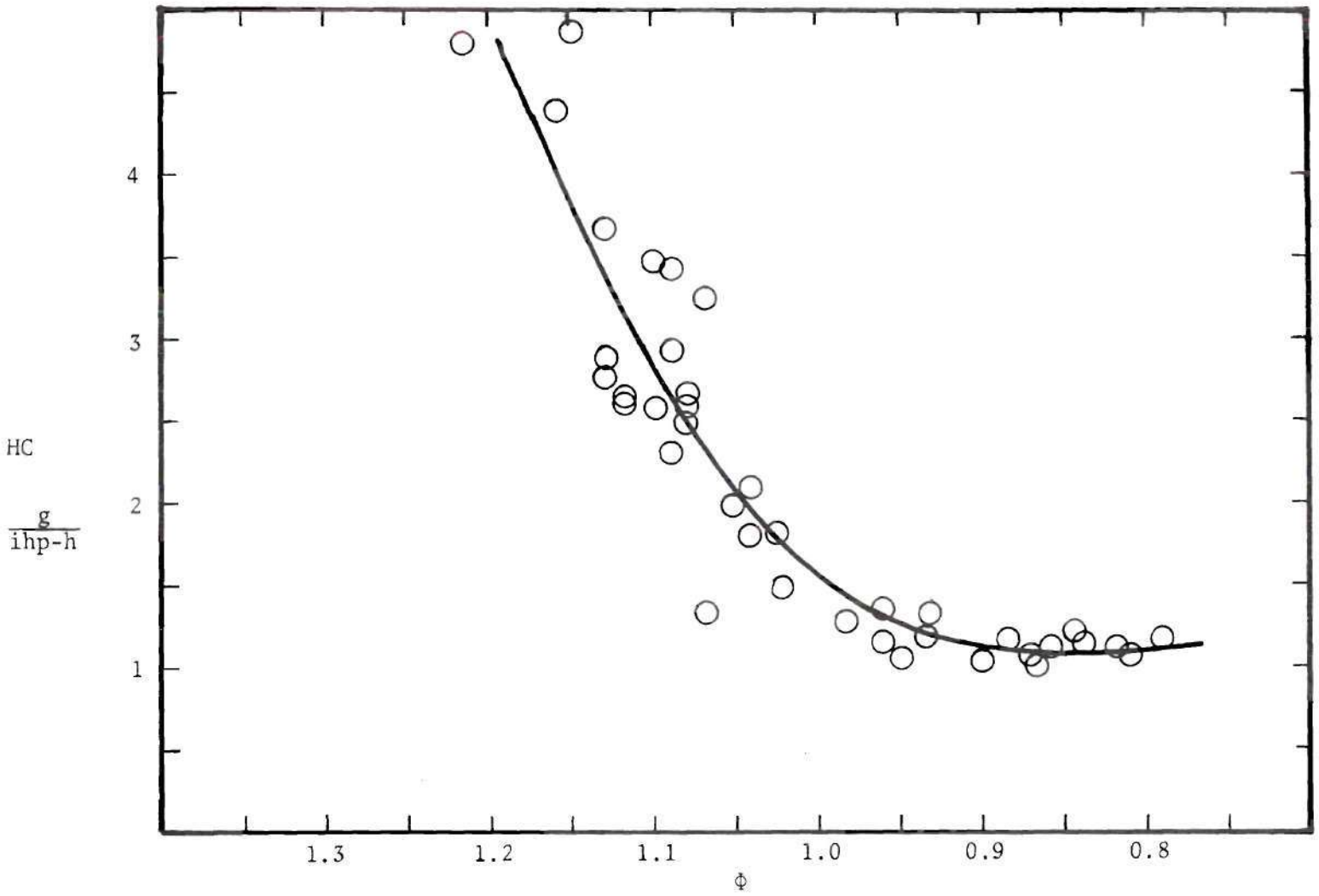

Figure 8. Ethanol HC Emissions at 1800 rpm vs. Equivalence Ratio

Emissions of unburned hydrocarbons while operating the engine on Indolene and ethanol are about the same for fuel-lean mixtures. However, emissions with ethanol appear to increase slightly more rapidly than those from Indolene as the equivalence ratio is increased. Emissions for each fuel are greater at the lower speed of 1000 rpm than at 1800 rpm. For equivalence ratios lower than 0.8, some increase in hydrocarbon emissions occurs, and this is probably due to engine misfire.

Emissions of NO_x

Mass emissions of oxides of nitrogen from the engine for the three fuels are shown in Figures 9-14. The characteristically steep slope of the oxide of nitrogen curve has made the data for NO_x less accurate than that for the other emissions. The data is also limited due to the fact that the maximum reading of the NO NDIR (4000 ppm) was exceeded on several occasions. This fact is substantiated by the emissions curves in Figures 9-14. While the peak values of NO_x emissions are difficult to assess, it can be readily ascertained that NO_x emissions with Indolene are substantially higher than those from either of the alcohols. For all fuels, peak NO_x emissions are found to occur around an equivalence ratio of 0.9.

Emissions of CO

Mass emissions of carbon monoxide from the engine for the three fuels are shown in Figures 15-20. For fuel-lean mixtures, carbon monoxide emissions were very low for all fuels and at both speeds. For increasingly fuel-rich mixtures from stoichiometric, carbon monoxide emissions from the engine while operating on Indolene seem

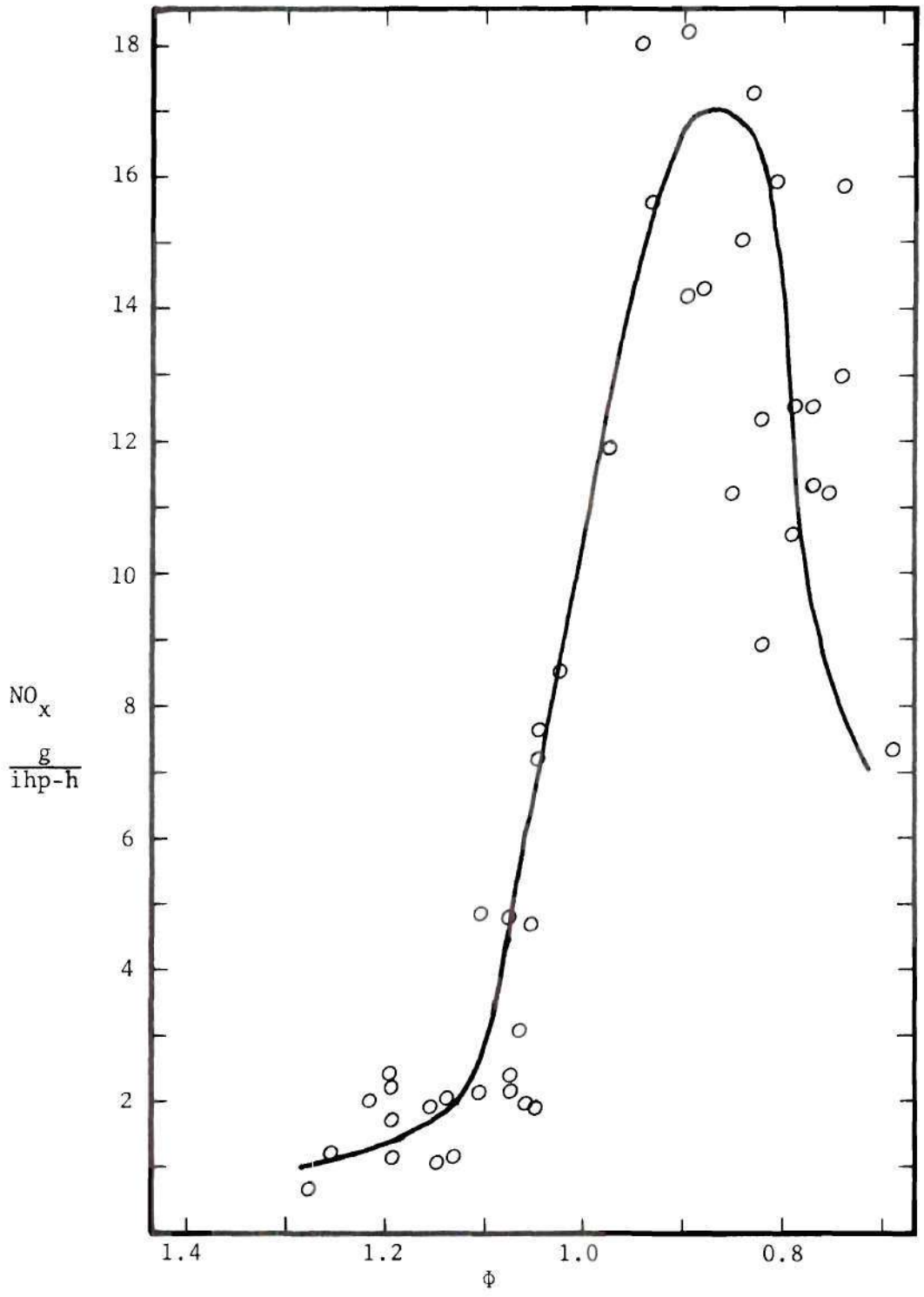

Figure 9. Indolene NO_x Emissions at 1000 rpm vs. Equivalence Ratio

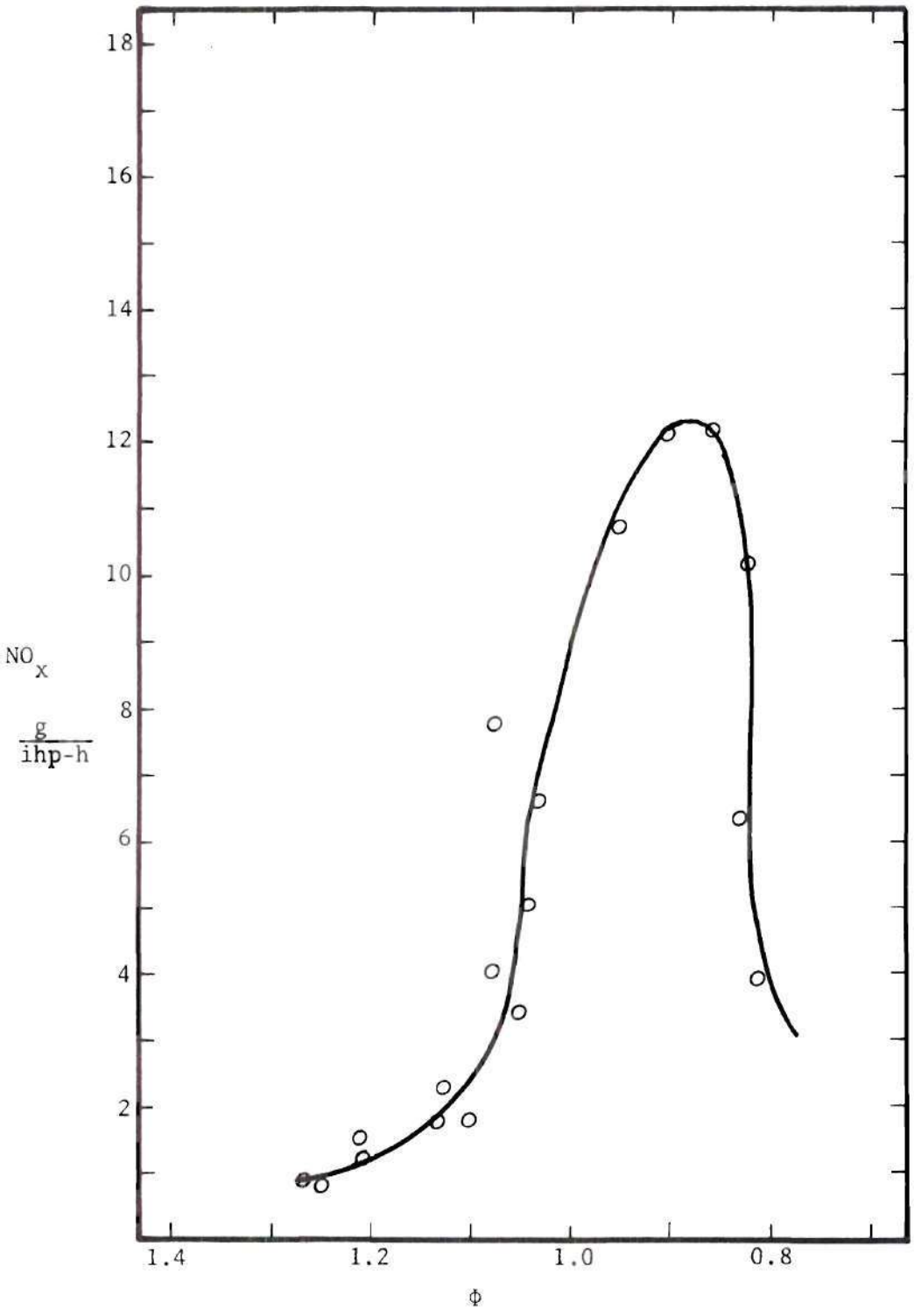

Figure 10. Indolene NO_x Emissions at 1800 rpm vs. Equivalence Ratio

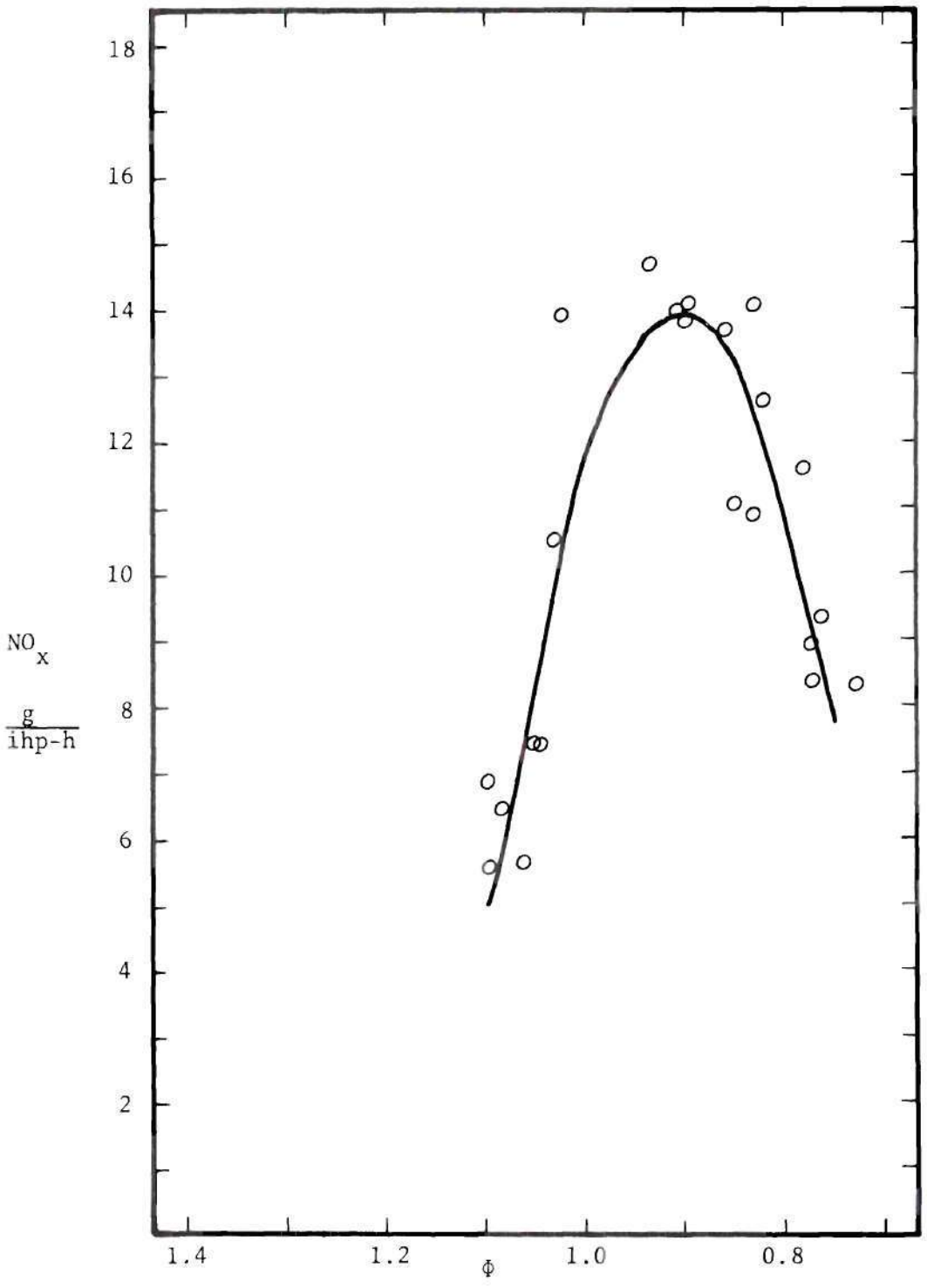

Figure 11. Methanol NO_x Emissions at 1000 rpm vs. Equivalence Ratio

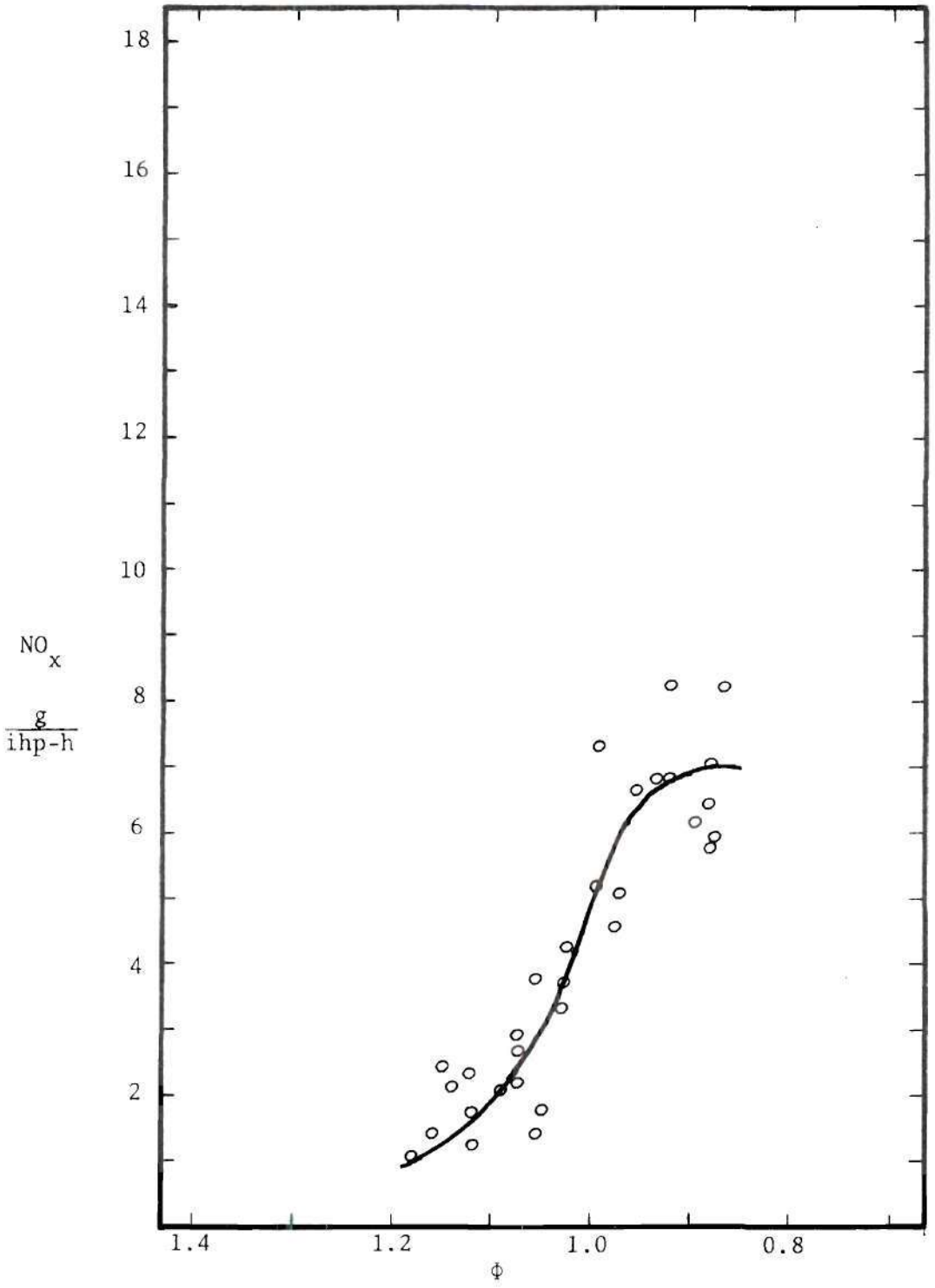

Figure 12. Methanol NO_x Emissions at 1800 rpm vs. Equivalence Ratio

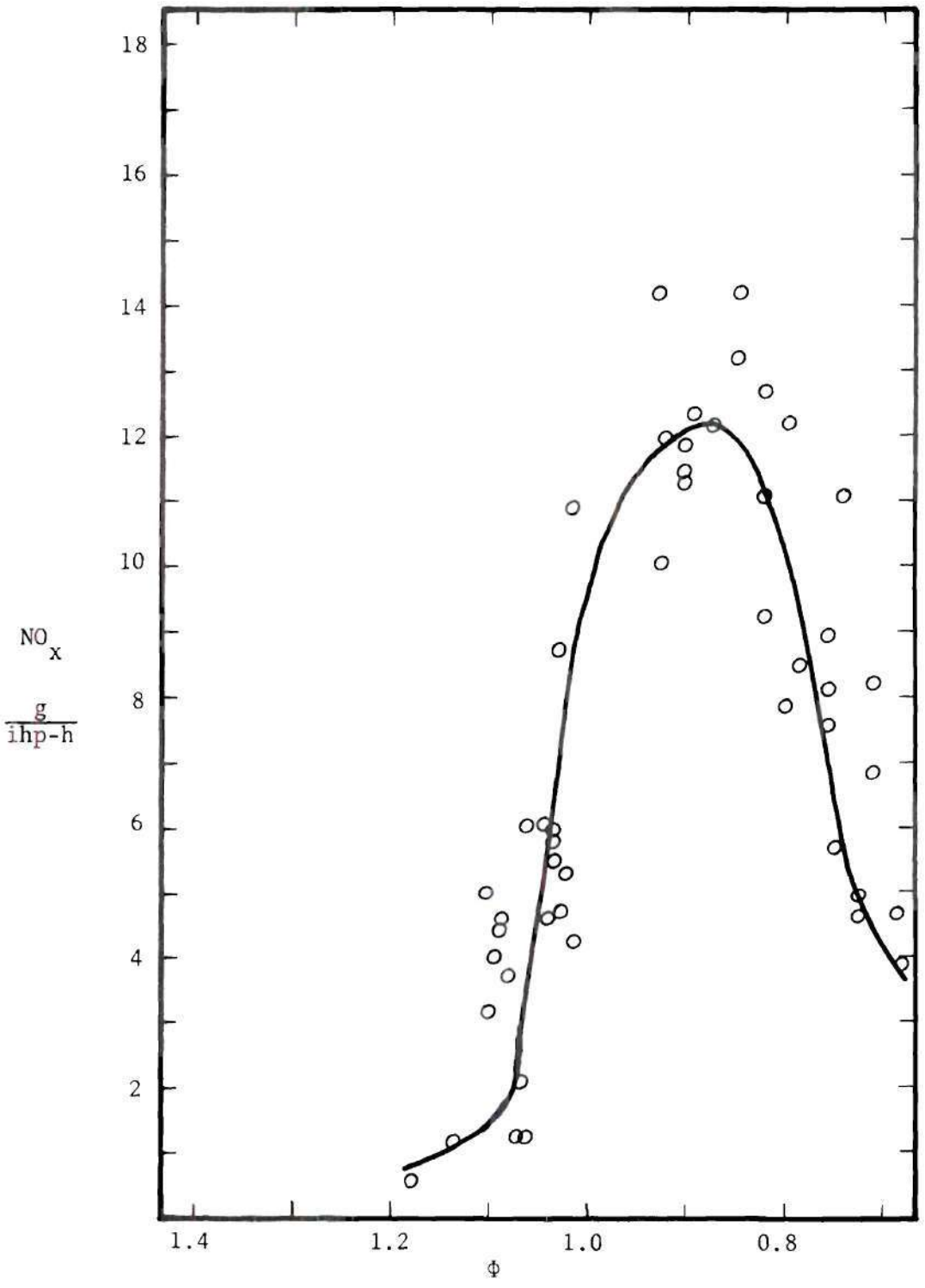

Figure 13. Ethanol NO_x Emissions at 1000 rpm vs. Equivalence Ratio

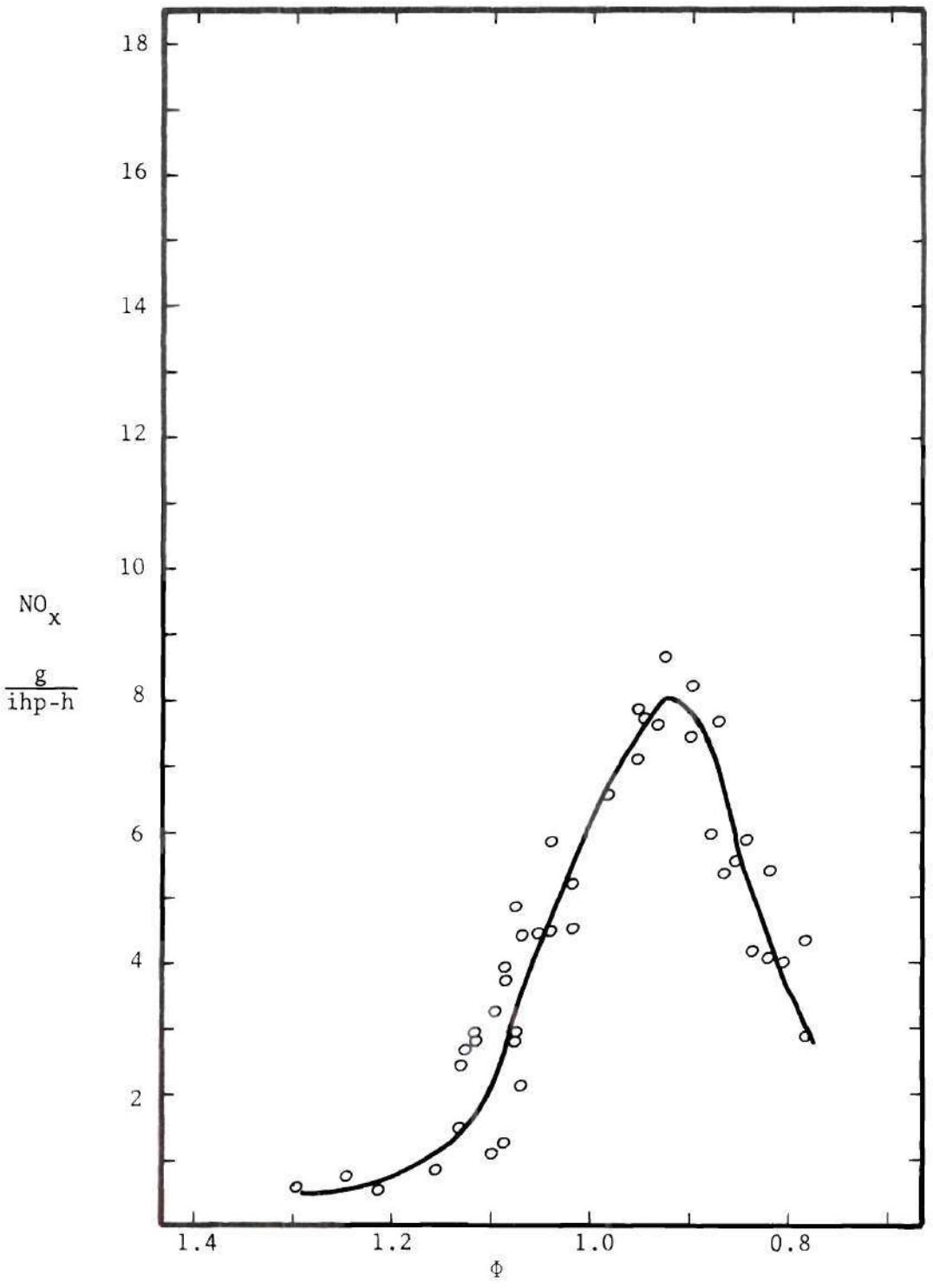

Figure 14. Ethanol NO_x Emissions at 1800 rpm vs. Equivalence Ratio

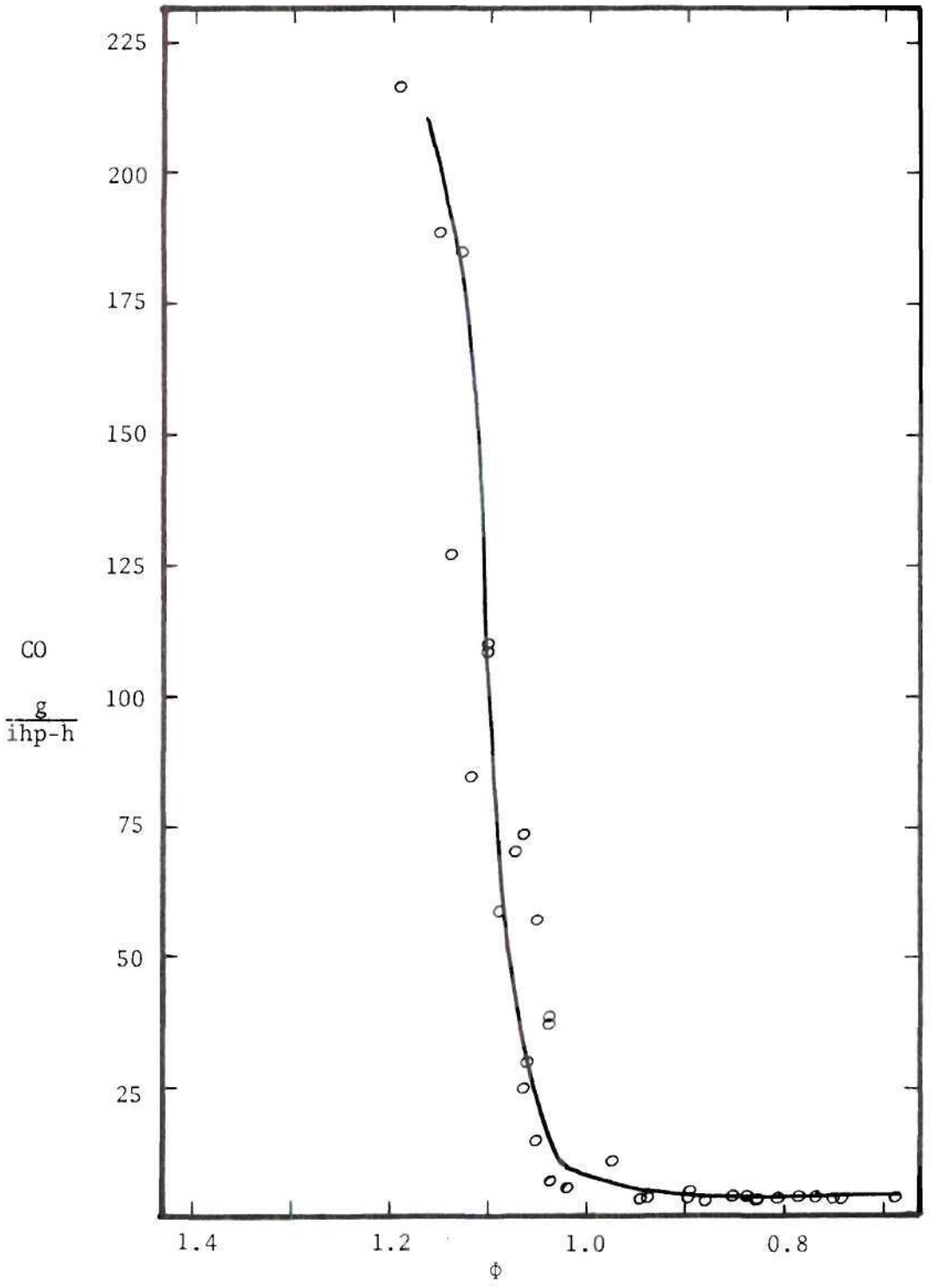

Figure 15. Indolene CO Emissions at 1000 rpm vs. Equivalence Ratio

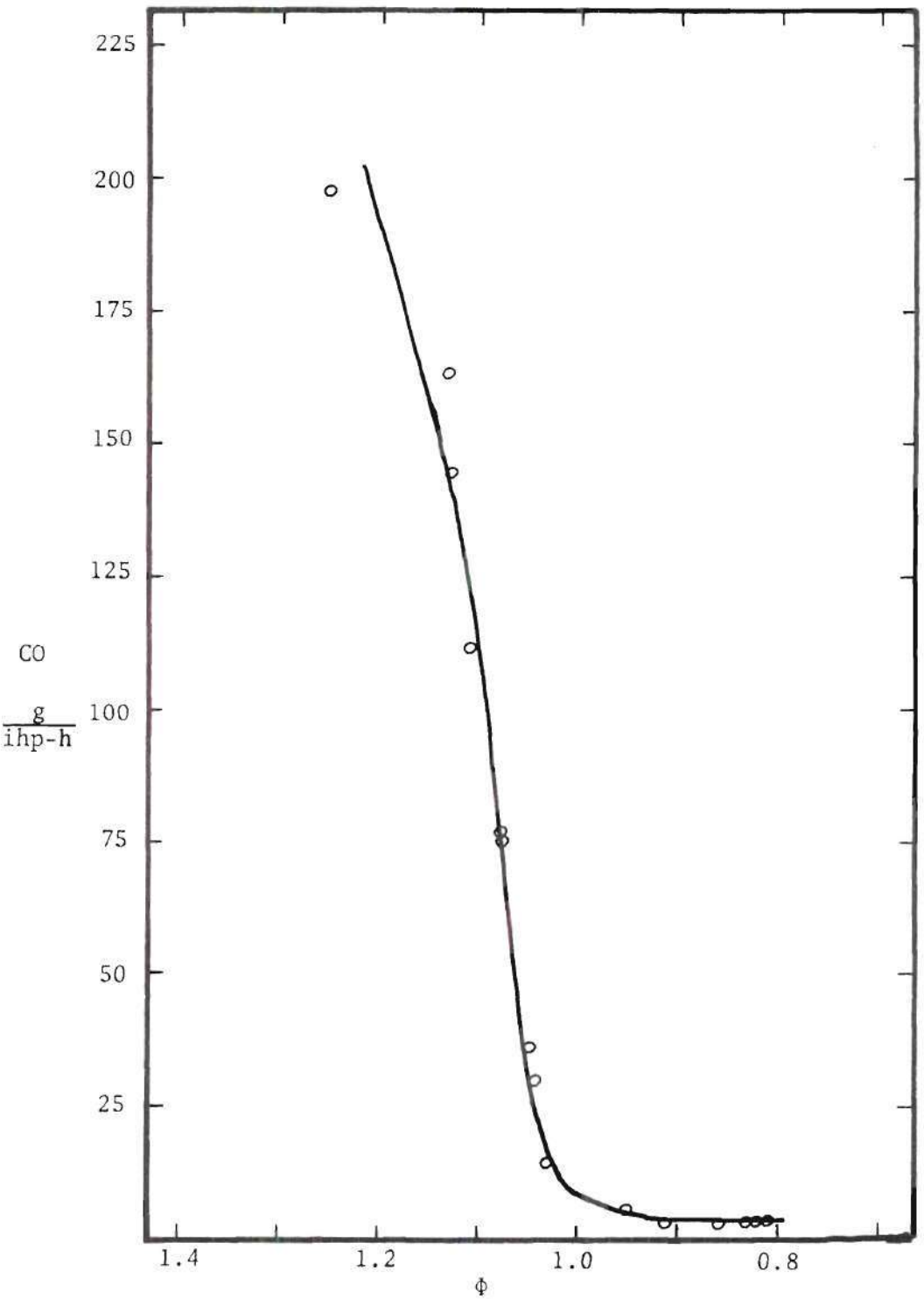

Figure 16. Indolene CO Emissions at 1800 rpm vs. Equivalence Ratio

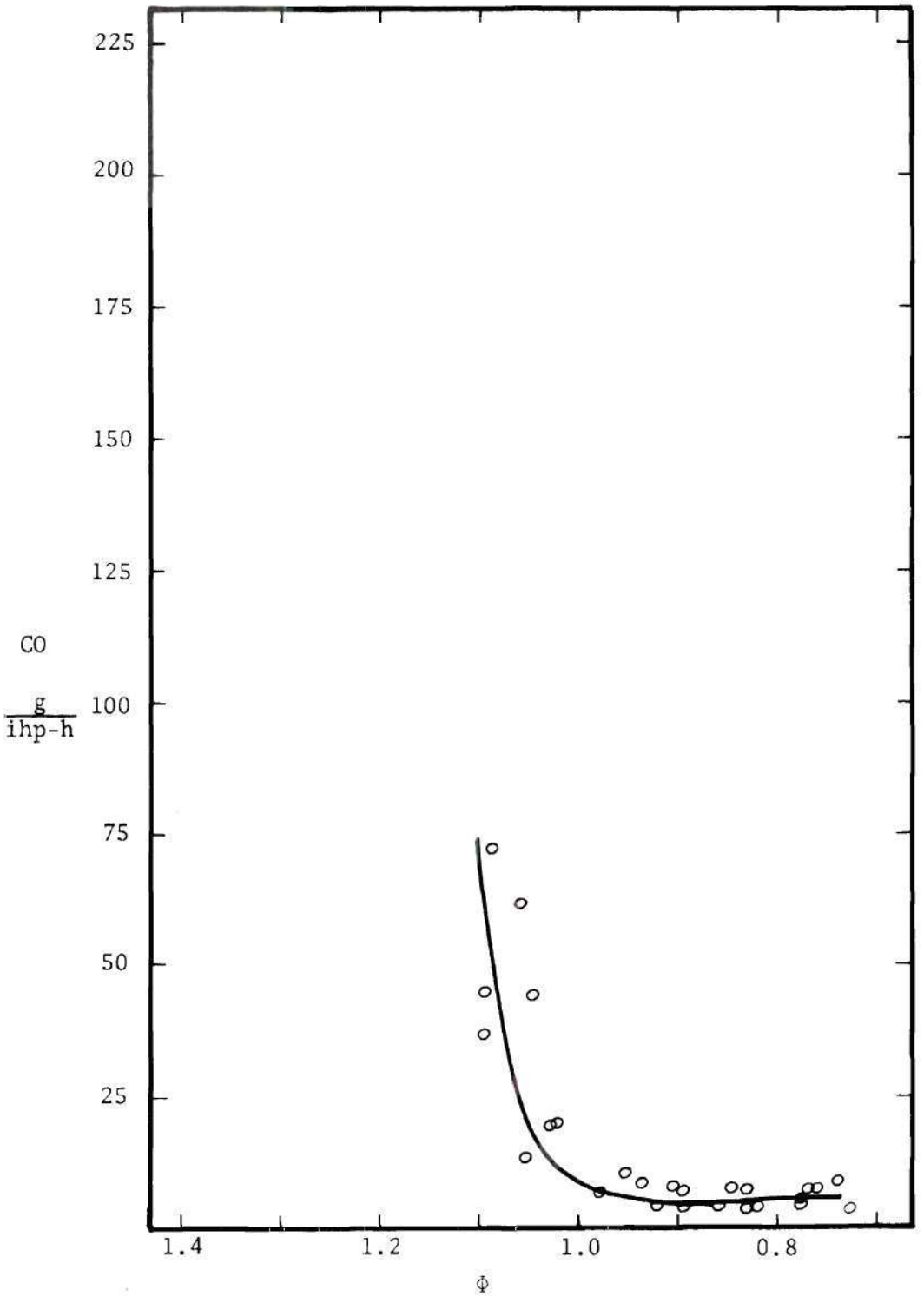

Figure 17. Methanol CO Emissions at 1000 rpm vs. Equivalence Ratio

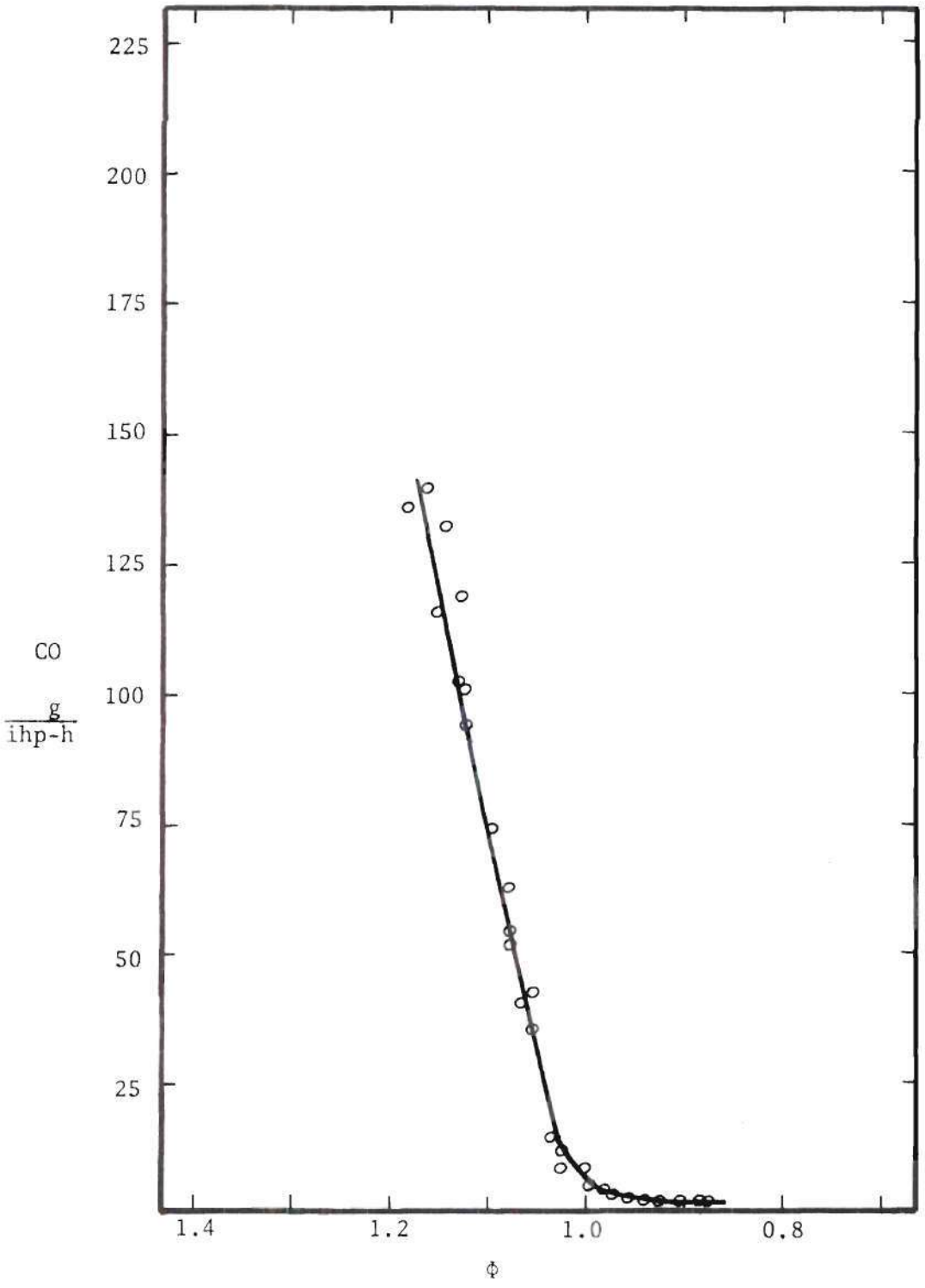

Figure 18. Methanol CO Emissions at 1800 rpm vs. Equivalence Ratio

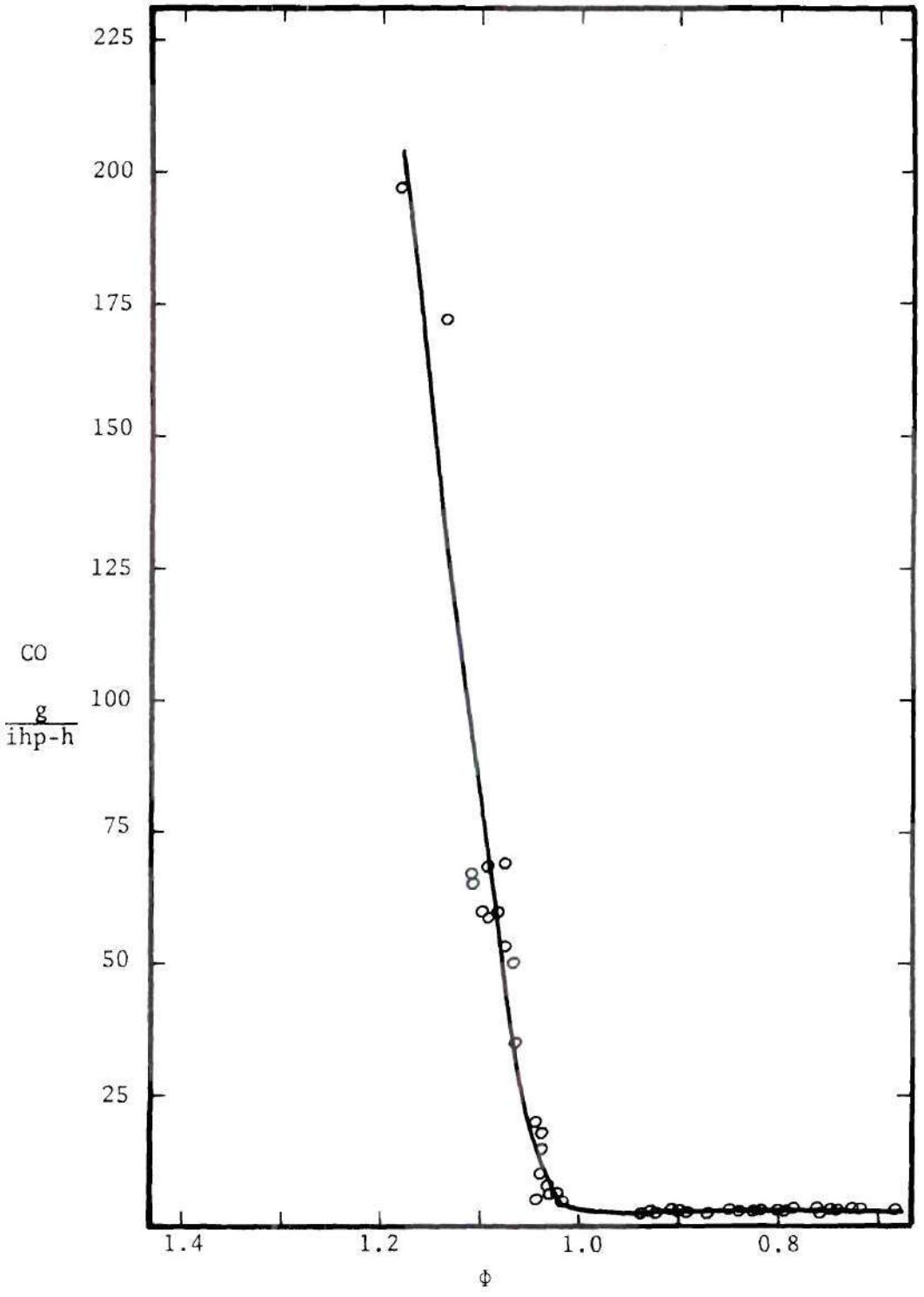

Figure 19. Ethanol CO Emissions at 1000 rpm vs. Equivalence Ratio

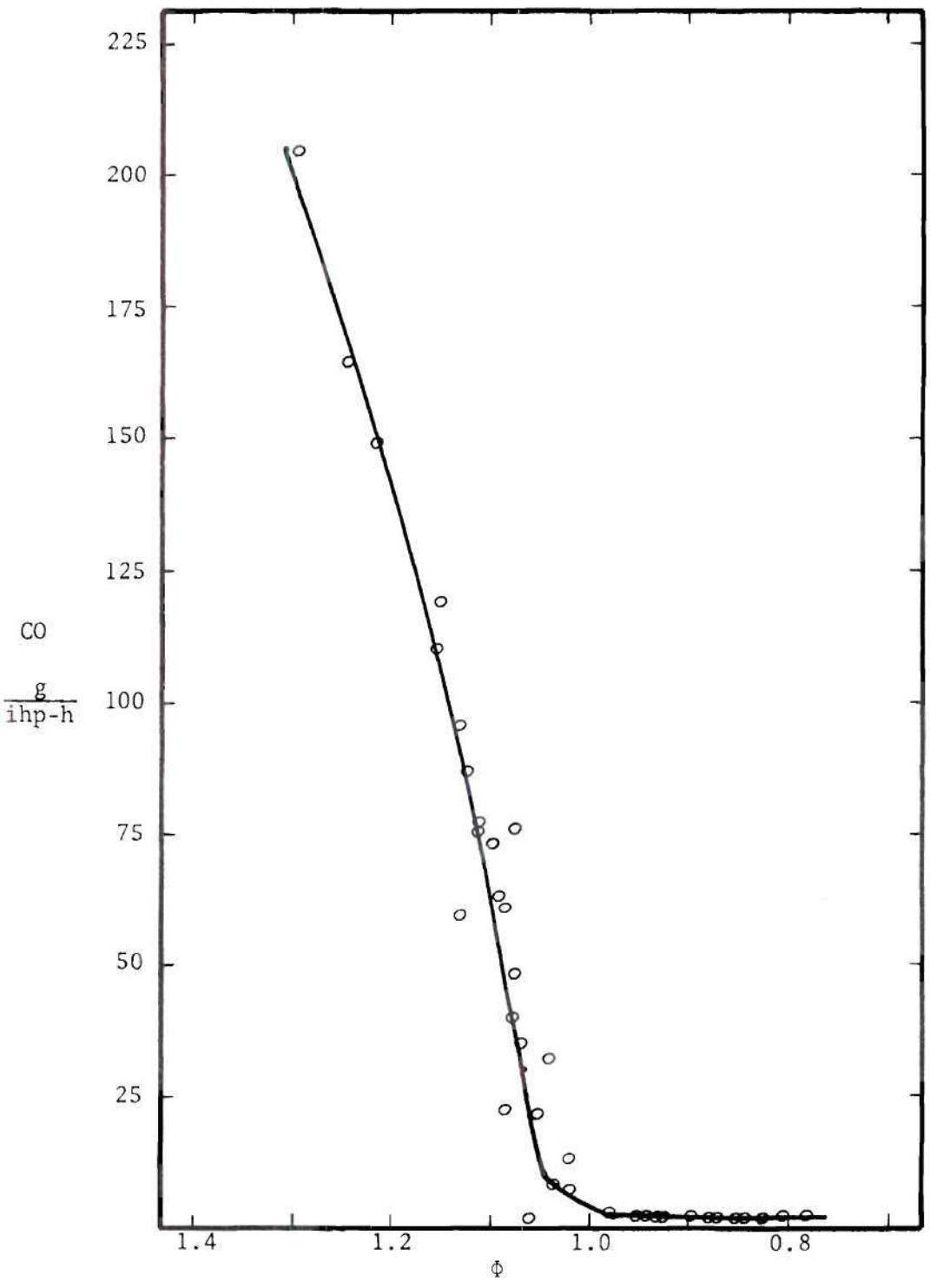

Figure 20. Ethanol CO Emissions at 1800 rpm vs. Equivalence Ratio

to increase slightly faster than those from methanol and ethanol. All carbon monoxide emissions data represents a marked dependence upon the equivalence ratio alone, a fact that has been well substantiated for pure hydrocarbon fuels. This appears to be true for alcohol fuels also.

Emissions of CO_2

Emissions of carbon dioxide are shown in table form in Appendix E rather than in figures. The shape of these curves is very similar to the curves for dry volume percentages of carbon dioxide contained in Appendix C. Approximate peak carbon dioxide values for each fuel and rpm are available from Appendix E.

Power Output

Figures 21, 22, and 23 depict the power output for each fuel at 1000 and 1800 rpm. The power output for all fuels is very similar. Brake horsepower values at 1000 rpm peak at approximately 2.6-2.8 lbs, and at approximately 0.9-1.1 lb at 1800 rpm.

Fuel Economy

Figures 24-29 display the indicated specific energy consumption for each fuel at 1000 and 1800 rpm versus equivalence ratio. Since power output has already been shown to be similar for all fuels, Figures 24-29 indicate that approximately 2.1 times as much methanol fuel as Indolene fuel must be consumed for identical power output. Ethanol requires approximately 1.6 times more fuel than Indolene for the same power output. These fuel consumption ratios hold at both

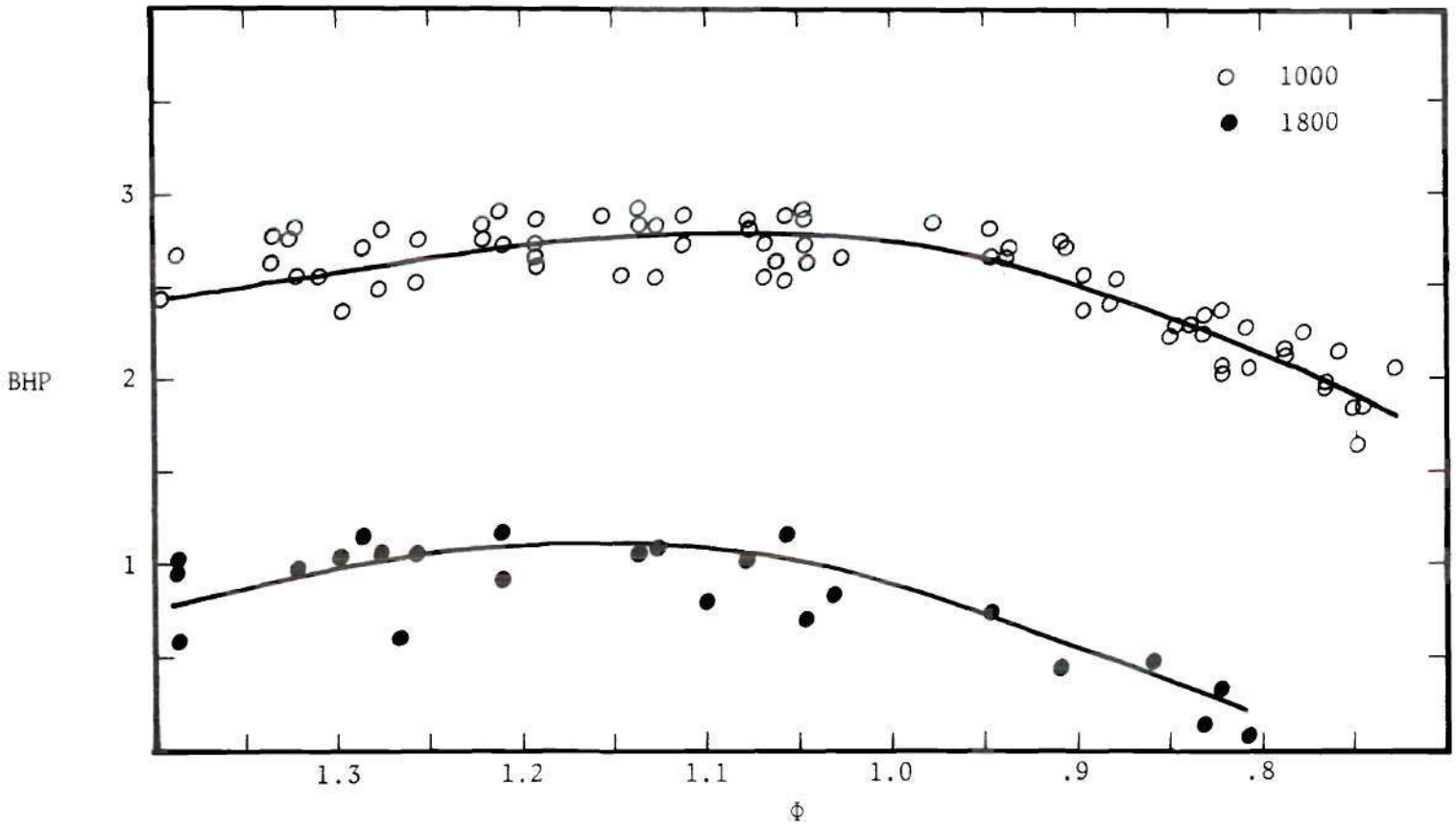

Figure 21. Indolene BHP at 1000 and 1800 rpm vs. Equivalence Ratio

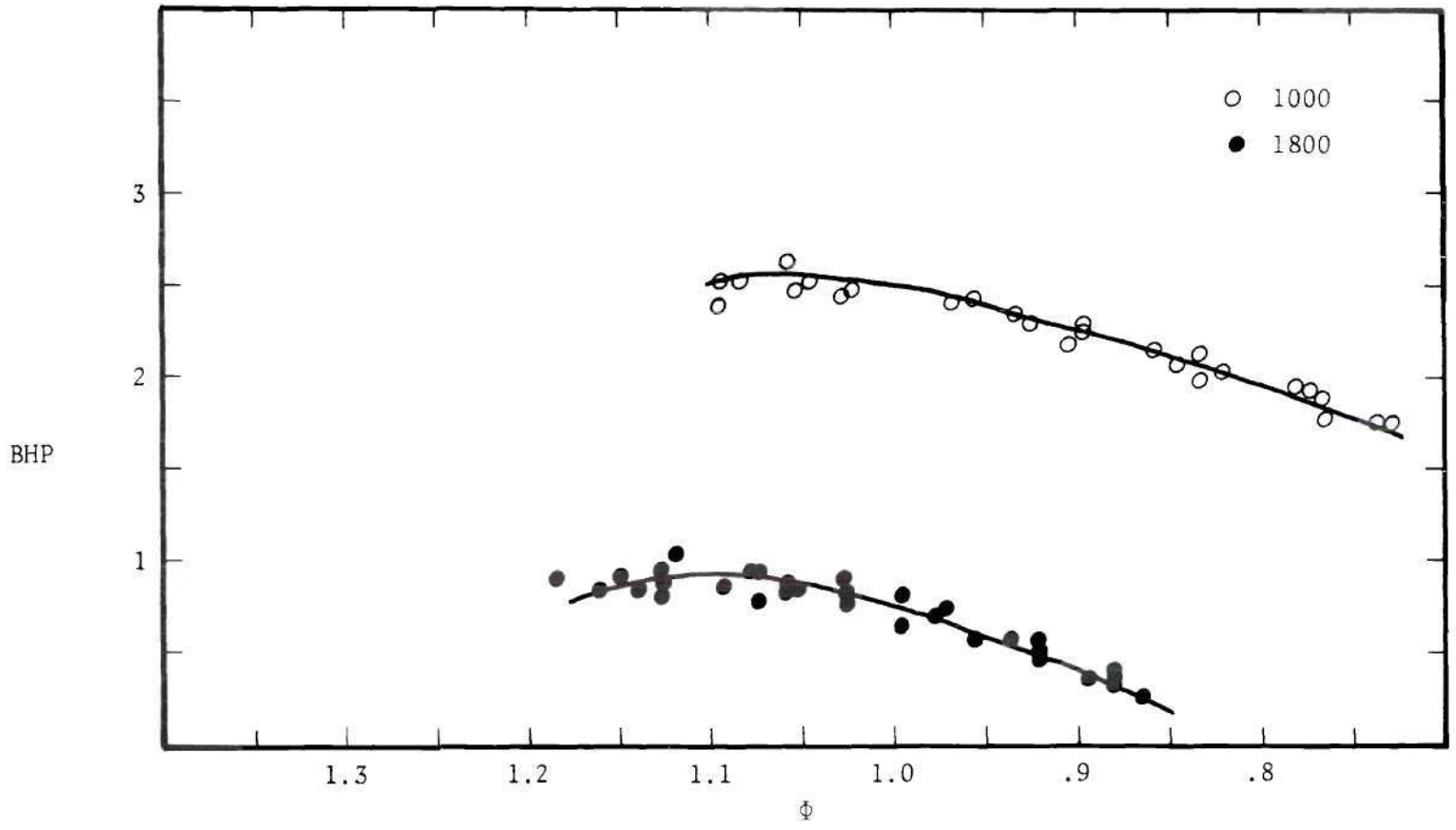

Figure 22. Methanol BHP at 1000 and 1800 rpm vs. Equivalence Ratio

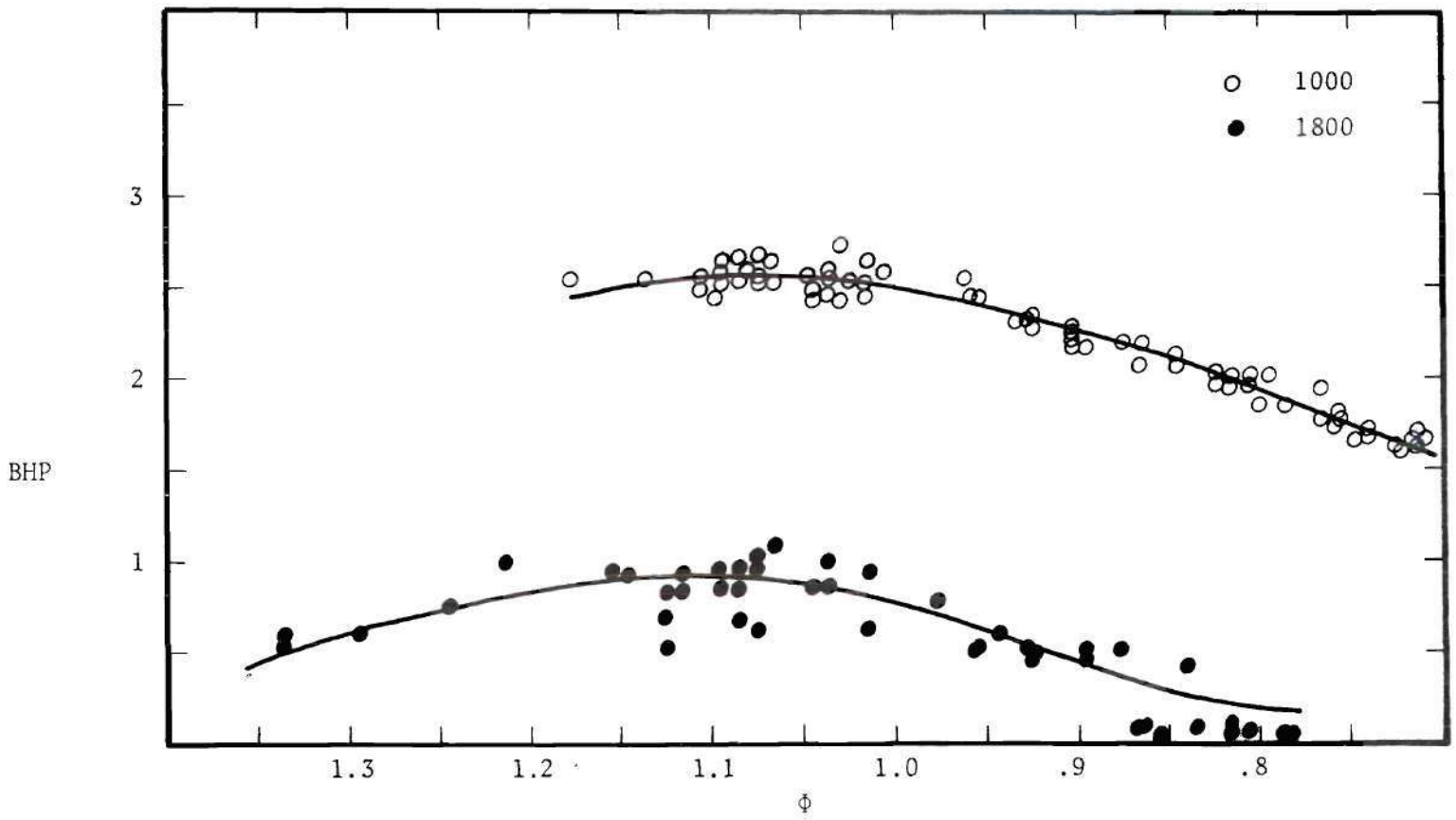

Figure 23. Ethanol BHP at 1000 and 1800 rpm vs. Equivalence Ratio

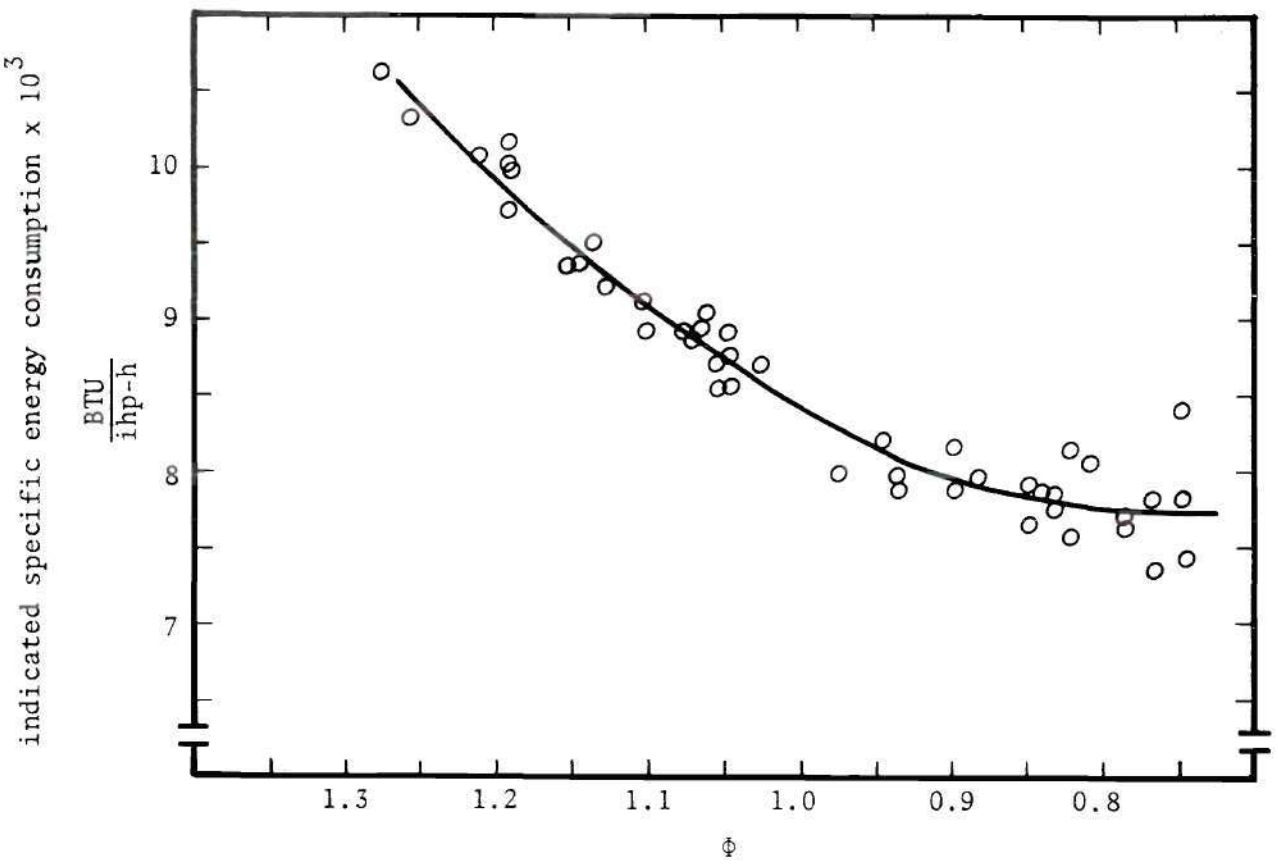

Figure 24. Indicated Specific Energy Consumption for Indolene at 1000 rpm vs. Equivalence Ratio

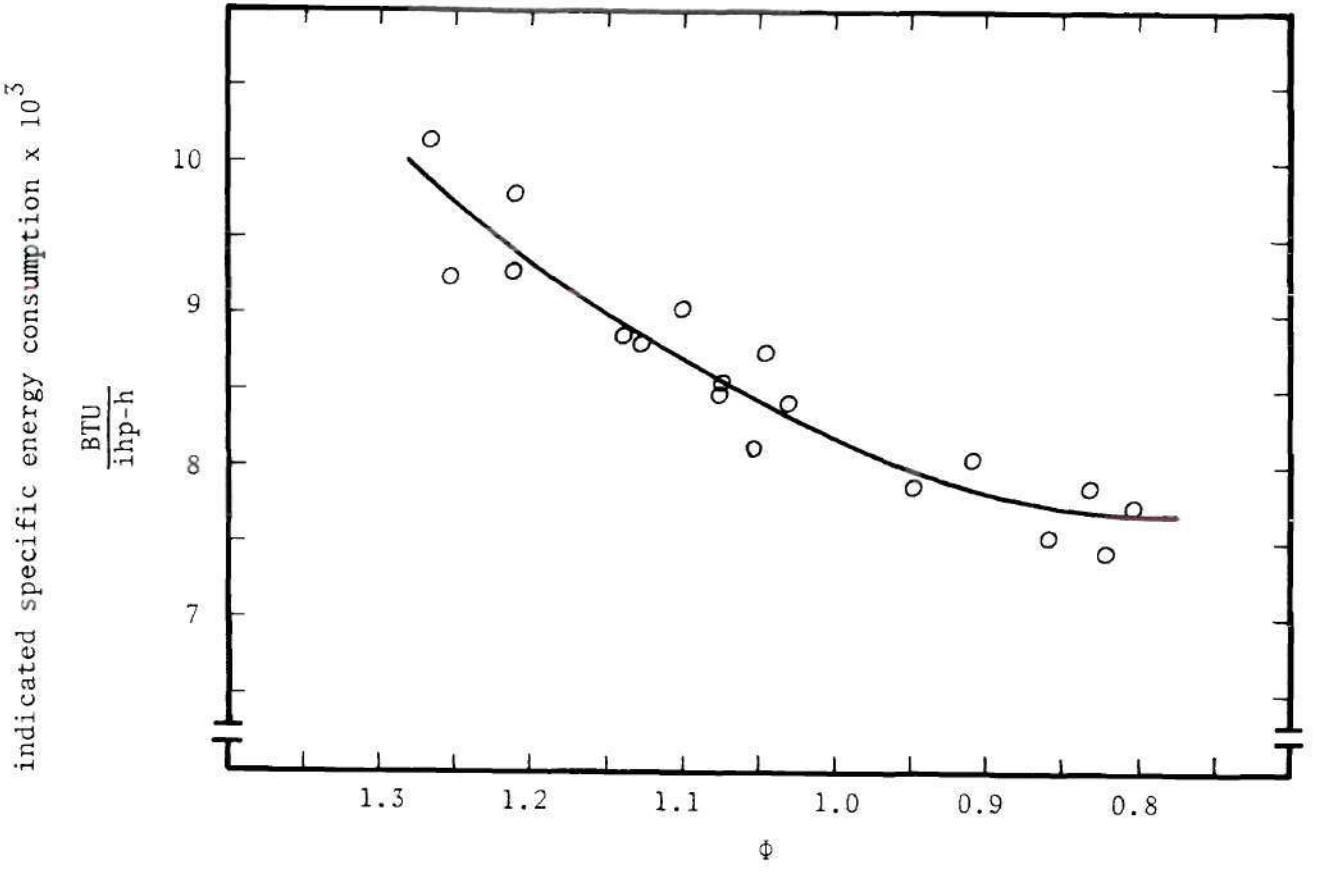

Figure 25. Indicated Specific Energy Consumption for Indolene at 1800 rpm vs. Equivalence Ratio

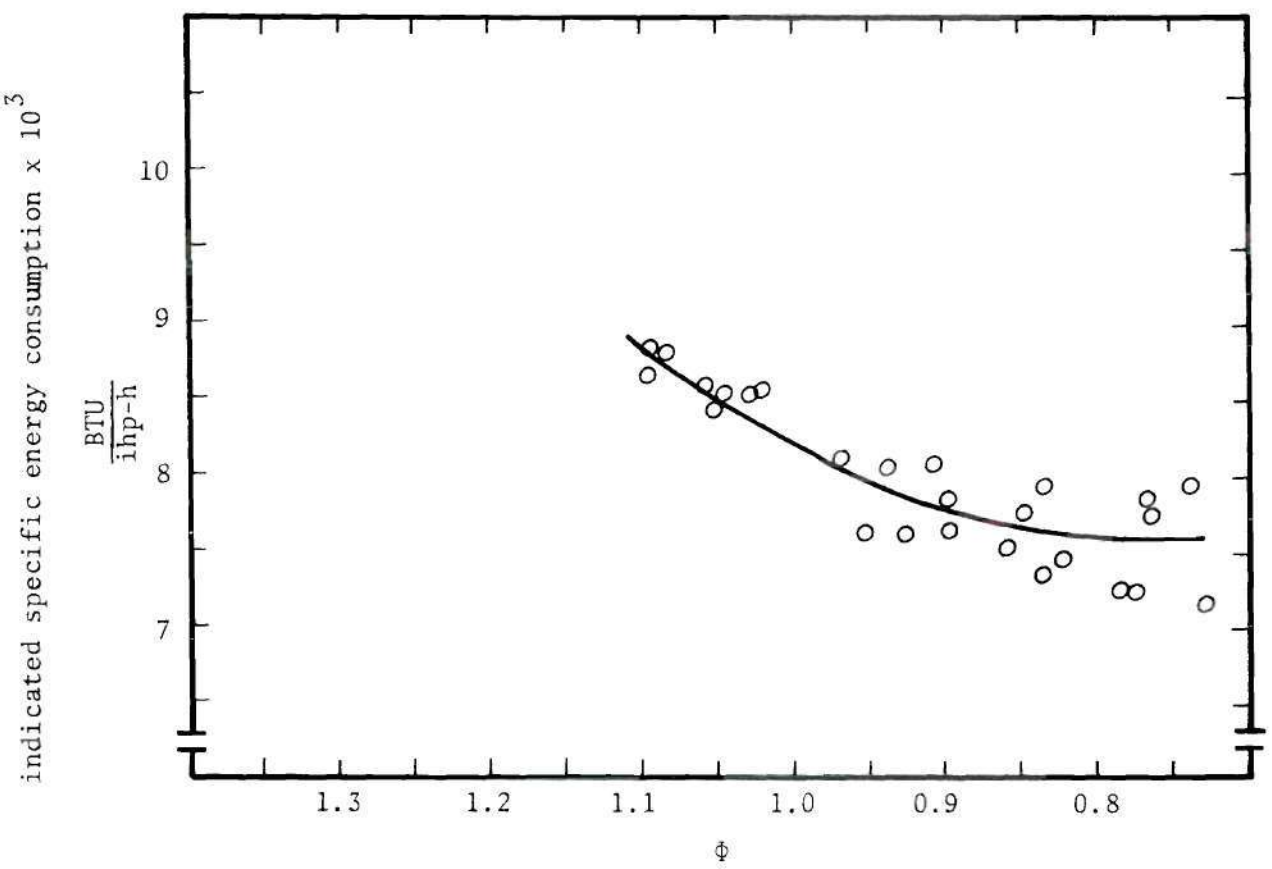

Figure 26. Indicated Specific Energy Consumption for Methanol at 1000 rpm vs. Equivalence Ratio

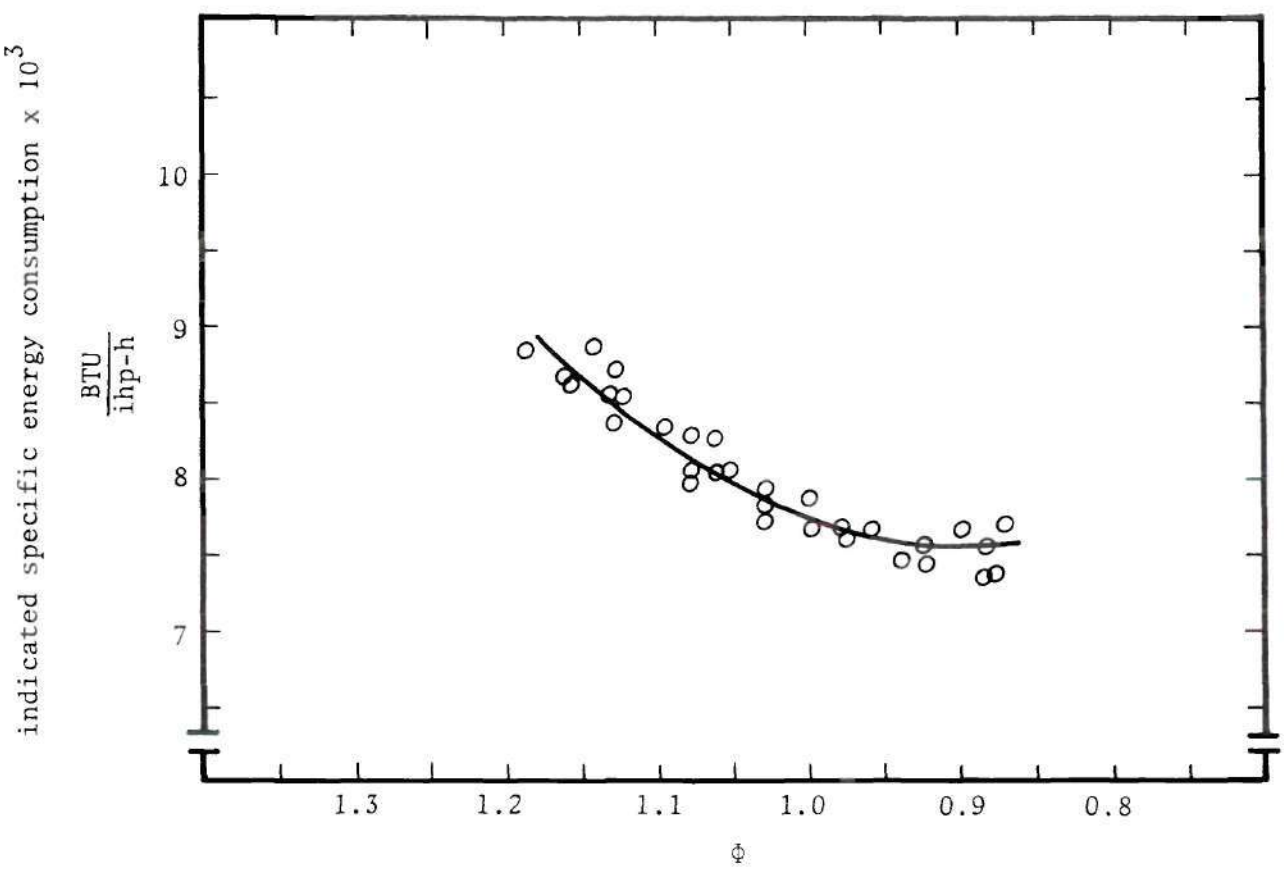

Figure 27. Indicated Specific Energy Consumption for Methanol at 1800 rpm vs. Equivalence Ratio

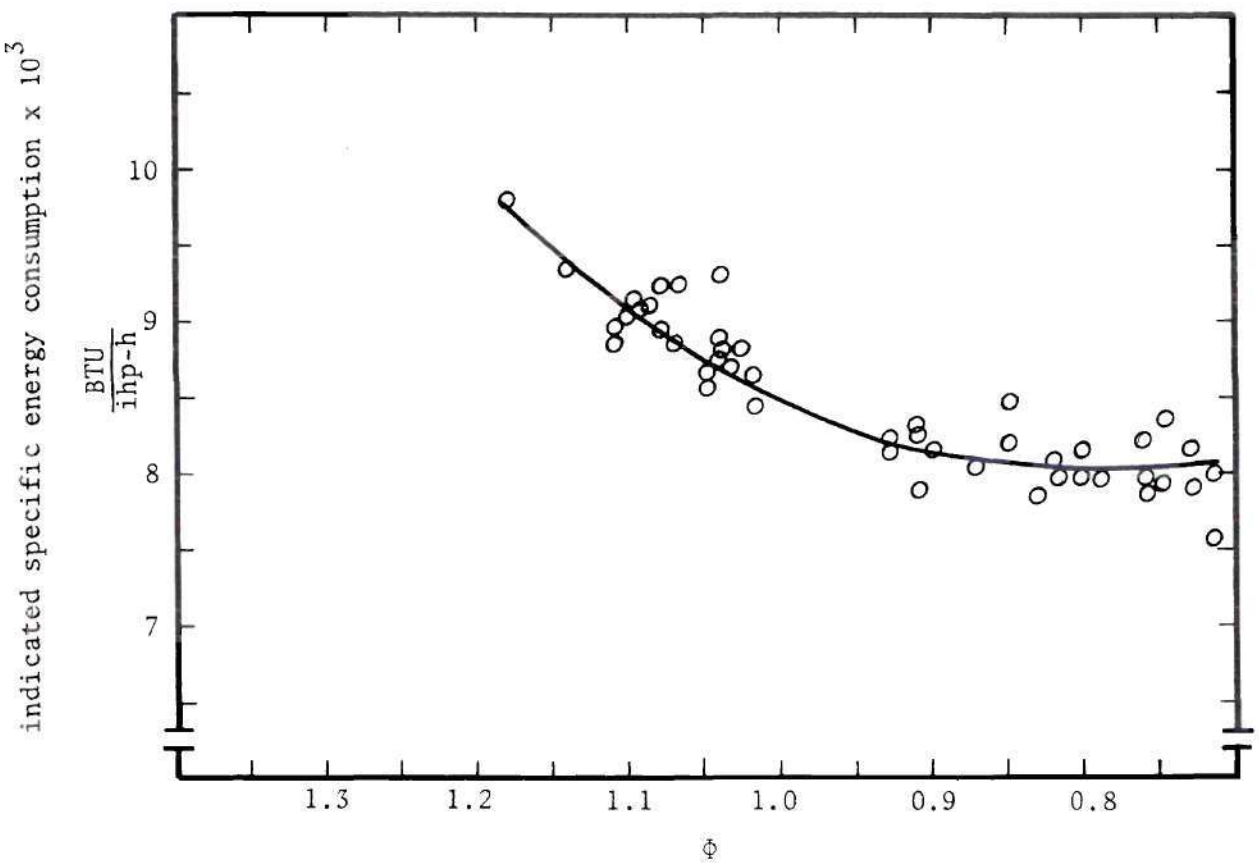

Figure 28. Indicated Specific Energy Consumption for Ethanol at 1000 rpm vs. Equivalence Ratio

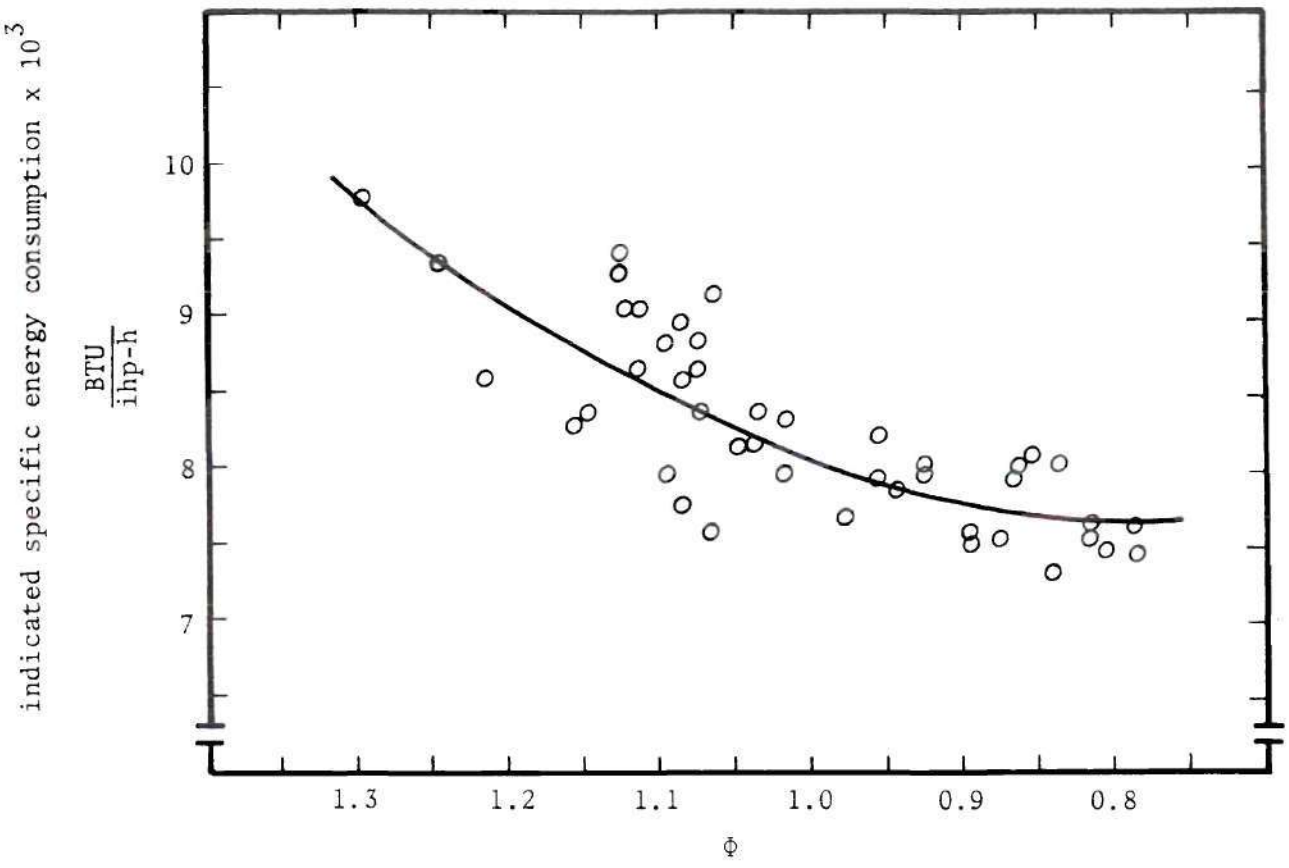

Figure 29. Indicated Specific Energy Consumption for Ethanol at 1800 rpm vs. Equivalence Ratio

engine speeds. The difference in the fuel consumption of the three fuel results directly from the difference in heating values of the fuels (see Table 2). The ratio of fuel consumption is identical to the ratio of the heating values of the fuels.

CHAPTER V

CONCLUSIONS

This study has established the engine power output, fuel consumption, and exhaust emission characteristics of a single cylinder spark ignition engine operated at maximum power spark advance with three fuels, Indolene, methanol, and ethanol. Comparison of these characteristics for the three fuels at two speeds allows the statement of the following conclusions.

1. Hydrocarbon mass emissions with methanol are significantly lower than with either Indolene or ethanol.
2. Oxides of nitrogen are lower for both alcohol fuels than for Indolene.
3. Carbon monoxide emissions are very nearly the same for all fuels, and appear to be a function of equivalence ratio only.
4. Power output is almost identical for all fuels.
5. The same power output is obtained with fuel consumption for methanol approximately 2.1 times greater than Indolene. The fuel consumption for ethanol is about 1.60 times greater than Indolene.

Because of the highly erratic behavior of the oxygen analyzer, it was not possible to make a valid comparative analysis of the air-fuel ratios computed by the D'Alleva and Spindt methods. It will suffice to say that for all fuels, when the meter appeared to be

working properly, the Spindt and D'Alleva methods for computing air-fuel ratios agreed quite well.

CHAPTER VI

RECOMMENDATIONS

There are many areas into which this work may be expanded. Specific subjects which merit more detailed investigation include variation of spark timing from maximum power spark advance, variation of compression ratio, use of other alcohol or alcohol-gasoline blends, and the introduction of a throttling device into the fuel system. An interesting project would be the measurement of exhaust emissions with these same three fuels using the CFR engine outfitted for fuel injection.

The CFR engine has already been shown to be a convenient and flexible apparatus in exhaust emission testing. The modification of the engine for this project may continually be improved upon in future experiments in obtaining even more extensive data.

Before the question of feasibility of alcohol use as a motor fuel can begin to be answered, much more elaborate and detailed experiments need to be conducted. The parameters discussed above will provide an excellent beginning in seeking the answers to some essential questions.

APPENDIX A

PROCEDURE FOR DETERMINING EFFECTIVE CARBON NUMBERS FOR METHANOL AND ETHANOL

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PROCEDURE FOR DETERMINING EFFECTIVE CARBON NUMBERS FOR METHANOL AND ETHANOL

Since the FID does not actually "see" every carbon atom of the alcohol molecule, it became necessary to experimentally determine the proportion of carbon atoms in methanol and ethanol that actually were counted. This ratio is called the effective carbon number (ECN), and is defined as the instrument response caused by a carbon atom in an alcohol molecule divided by the instrument response caused by an aliphatic carbon atom. In other words, the effective carbon number is the meter reading in ppmC divided by the actual ppmC.

The general procedure by which the effective carbon numbers for the alcohols were found is as follows:

1. Select a flask of known capacity, so that the mass of air in the flask may be calculated from the ideal gas law.
2. Inject a known quantity of fuel, and allow the fuel to evaporate and attain thermal equilibrium within the flask.
3. Calculate partial pressures of air and fuel in the flask.
4. The parts per million (ppm) of fuel in the flask is the ratio of the partial pressure of fuel to the partial pressure of air.
5. Direct the air-fuel vapor through the FID and record the meter reading in ppmC.
6. The effective carbon number for this particular fuel is

the meter reading in ppmC divided by the actual ppmC in the flask from step 4.

A round five-liter glass flask was modified for this experiment by attaching 4-mm teflon stopcocks at opposite poles of the flask, and attaching a nipple covered by a rubber septum at the center. The exact capacity of the flask was determined by filling with distilled water and weighing to the nearest 0.01 lb.

Prior to every run, the flask was rinsed thoroughly with distilled water, and compressed air was run through the flask until the interior was completely dry. The flask was then filled to a slight over pressure with the compressed air, connected to the FID, and the air was run through the FID. This gave a background count of carbon atoms present in the flask. Usually this count was of the order of two to three parts per million. If the count was appreciably higher, the flask was flushed again with distilled water, and the entire process repeated.

Once a suitably low background count was obtained, the flask was blown out again with compressed air and refilled to a slight over pressure. This time the pressure was recorded with a pressure gauge accurate to one part in 1500. A known amount of fuel (5 microliters) was injected through the septum with a Hamilton 701N Syringe. This syringe is designed to deliver liquids of up to 10 microliters at an accuracy of $\pm$1%. The fuel was allowed to evaporate and achieve thermal equilibrium. Generally, several hours were sufficient. At least one test for each fuel was conducted overnight, with consistent results and no appreciable leakage of vapor.

After the air-fuel vapor was believed to be in thermal equilibrium, the flask was connected to the sample train upstream from the pump. The pump was turned on with the stopcocks still closed to clear the sample lines as much as possible. One stopcock was then opened and the pump pulled the air-fuel vapor from the flask and pumped it into the FID. The maximum meter reading was noted.

Sufficient data were now available to calculate the proportion of carbon atoms actually recorded by the hydrocarbon analyzer. The above procedure was repeated at least three times for each alcohol fuel, and the average of the effective carbon numbers was used.

Sample Calculation

Fuel: Methanol

Specific gravity = 0.792

Volume of flask = 5.02 liters

Ambient pressure = 14.30 psia

Ambient temperature = 76 F = 297.6 K

Flask pressure = 17.32 psia

Amount of fuel injected = 5 microliters

Meter reading = 380 ppm

Step 1:

Calculate the mass of air in flask from ideal gas law:

$$m_a = 7.024 \text{ gm}$$

Step 2:

Calculate mass of fuel injected into flask:

$$m_i = 3.96 \times 10^{-4} \text{ gm}$$

Step 3: Calculate the partial pressures of air and fuel from the ideal gas law:

$$p_a = 1.178 \text{ psi}$$

$$p_i = 6.02 \times 10^{-4} \text{ psi}$$

Step 4: The ratio of partial pressures is the actual number of carbon atoms in the air-fuel vapor:

$$X_i = \frac{6.02 \times 10^{-4}}{1.178} = 511 \text{ ppm}$$

Step 5: The effective carbon number is the ratio of the meter reading to the actual ppmC of the air-fuel mixture:

$$ECN = \frac{380}{511} = .744 \quad .$$

APPENDIX B

SAMPLE DATA SHEET

APPENDIX B

SAMPLE DATA SHEET

Fuel: Ethanol
Date: 27 Mar 74
Fuel Jet: .035 in

	Run					
	1	2	3	4	5	6
Δ"	.46	.46	.46	.46	.46	.46
p_{oil}, psi	38	38	38	38	38	38
t_{oil}, F	138	138	138	137	136	136
t_{air}, F	119	119	120	120	120	120
fb	.40	.80	1.2	1.7	2.2	2.6
SA	19	19	19	19	19	19
t_m, F	139	139	142	142	143	142
t_w, F	210	210	210	210	210	210
#, lbs	15.00	14.85	13.80	12.25	10.80	9.60
HC, ppm	70	51	32	31	27	28.5
NO, ppm	.41	.49	.75	.85	.58	.31
CO_2, %	.745	.79	.77	.72	.66	.64
CO, %	.50	.15	.02	.02	.02	.02
O_2, %	4.6	4.6	5.4	7.0	8.4	9.2
RPM	1011 1011	1007 1008	1008 1005	1008 1009	983 986	994 1002
T_{room}, F	89	89	89	89	89	90

APPENDIX C

DERIVATION OF D'ALLEVA CHARTS FOR GENERAL FUEL MOLECULE OF FORM CH_yO_z

APPENDIX C

DERIVATION OF D'ALLEVA CHARTS FOR GENERAL FUEL MOLECULE OF FORM CH_yO_z

Reference (38) derives charts for exhaust emissions versus air-fuel ratios for fuel molecules containing various ratios of carbon and hydrogen atoms only. We wish to establish similar charts so that they may be applied to a more general fuel molecule, CH_yO_z, and therefore, also be applied to alcohol fuels.

The general chemical reaction, following a similar argument to that of D'Alleva, is assumed to be:

$$CH_yO_z + nO_2 + 3.76nN_2 = a\ CO_2 + b\ H_2O + (1-a-p)\ CO + \tfrac{1}{2}(y-2b-4p)H_2 + p\ CH_4 + 3.76nN_2 + \tfrac{1}{2}(z+2n-a-b-1+p)\ O_2 \quad (1)$$

If we assume that the number of moles of CH_4 in the exhaust is zero, i.e., that no unburned hydrocarbons are present in the exhaust, the chemical reaction may be written:

$$CH_yO_z + nO_2 + 3.76nN_2 = aCO_2 + bH_2O + (1-a)CO + \tfrac{1}{2}(y-2b)H_2 + 3.76nN_2 + \tfrac{1}{2}(z+2n-a-b-1)O_2 \quad (2)$$

Separate formulas may be written from the general chemical reaction for both rich and lean air-fuel mixtures. For rich mixtures, assuming that no oxygen is present as a product, we may write the chemical

reaction equation as:

$$CH_yO_z + nO_2 + 3.76nN_2 = aCO_2 + bH_2O + (1-a)CO + 3.76nN_2$$

$$+ \tfrac{1}{2}(y-2b)H_2 \qquad (3)$$

The chemical reaction equation for lean mixtures, assuming that no CO is formed as a product, is written:

$$CH_yO_z + nO_2 + 3.76nN_2 = aCO_2 + bH_2O + 3.76nN_2 + \tfrac{1}{2}(z+2n-2a-b)O_2 \qquad (4)$$

This equation may be reduced to a simpler form by balancing the hydrogen and carbon atoms on both sides of the equation.

$$CH_yO_z + nO_2 + 3.76nN_2 = CO_2 + \frac{y}{2}H_2O + 3.76nN_2 + \tfrac{1}{2}(z+2n-2-\frac{y}{2})O_2 \qquad (5)$$

The water-gas equilibrium equation is expressed as:

$$K = \frac{(CO)(H_2O)}{(CO_2)(H_2)} = \frac{(1 - a)b}{a\left(\frac{y}{2} - b\right)} \qquad (6)$$

The water-gas equilibrium constant, K, is assumed by D'Alleva to be 3.8. An expression for the moles of products of CO_2 is obtained by rearranging the terms in equation (6).

$$a = \frac{2b}{Ky - 2Kb + 2b} \qquad (7)$$

Rich Mixtures

From equation (3), the rich reaction equation, the oxygen balance may be expressed:

$$z + 2n = 2a + b + 1 - a \tag{8}$$

Rearranging, and solving for n:

$$n = \tfrac{1}{2}(a + b + 1 - z) \tag{9}$$

Substituting the expression for a obtained from the water-gas equilibrium equation, we obtain:

$$n = \frac{1}{2}\left(\frac{2b}{Ky - 2Kb + 2b} + b + 1 - z\right) \tag{10}$$

Rearranging terms and solving for b:

$$0 = b^2(2 - 2K) + b(Ky - 2K + 4 + 2Kz - 2z - 4n + 4nK) + Ky - Kyz - 2nKy \tag{11}$$

$$b = \frac{-(Ky-2K+4+2Kz-2z-4n+4nK) - \sqrt{(Ky-2K+4+2Kz-2z-4n+4nK)^2 - 4(2-2K)(Ky-Kyz-2nKy)}}{2(2-2K)} \tag{12}$$

From equation (3), the total moles in dry exhaust is:

$$T = 3.76n + 1 + \frac{y}{2} - b \tag{13}$$

Consequently, exhaust gas concentration for particular emittants become:

$$(XCO_2) = \frac{a}{3.76n + 1 + \frac{y}{2} - b} = \frac{2n - b - 1 + z}{3.76n + 1 + \frac{y}{2} - b} \tag{14}$$

$$(XCO) = \frac{1 - a}{3.76n + 1 + \frac{y}{2} - b} = \frac{2 - 2n + b - z}{3.76n + 1 + \frac{y}{2} - b} \tag{15}$$

Lean Mixtures

Similarly for lean mixtures, the total moles in dry exhaust is given in equation (5) as:

$$\text{total moles} = 4.76n + \frac{z}{2} - \frac{y}{4} \tag{16}$$

The exhaust gas concentrations for lean mixtures are then given by:

$$(XCO_2) = \frac{1}{4.76n + \frac{z}{2} - \frac{y}{4}} \tag{17}$$

$$(XCO) = \frac{\frac{1}{2}(z + 2n - 2 - \frac{y}{2})}{4.76n + \frac{z}{2} - \frac{y}{4}} \tag{18}$$

Since the general fuel molecule has a molecular weight equal to (12 + y + 16z), the air-fuel ratio may be expressed as follows:

$$AF = \frac{(32n + 28(3.76)n)}{12 + y + 16z} = \frac{137.39n}{12 + y + 16z} \tag{19}$$

Substituting equation (19) into equations (14), (15), (17), and (18) will provide exhaust concentrations as a function of air-fuel ratio and fuel molecular fuel composition. The type of fuel used will uniquely determine values for y and z, and we then obtain expressions for exhaust concentrations in terms of air-fuel ratios only. These expressions for exhaust concentrations are then easily converted to

functions of equivalence ratio alone. D'Alleva charts were thus derived for Indolene ($y = 1.86$, $z = 0$), methanol ($y = 4$, $z = 1$), and ethanol ($y = 3$, $z = 1/2$). The D'Alleva charts for these three fuels are shown in Figures 30, 31, and 32.

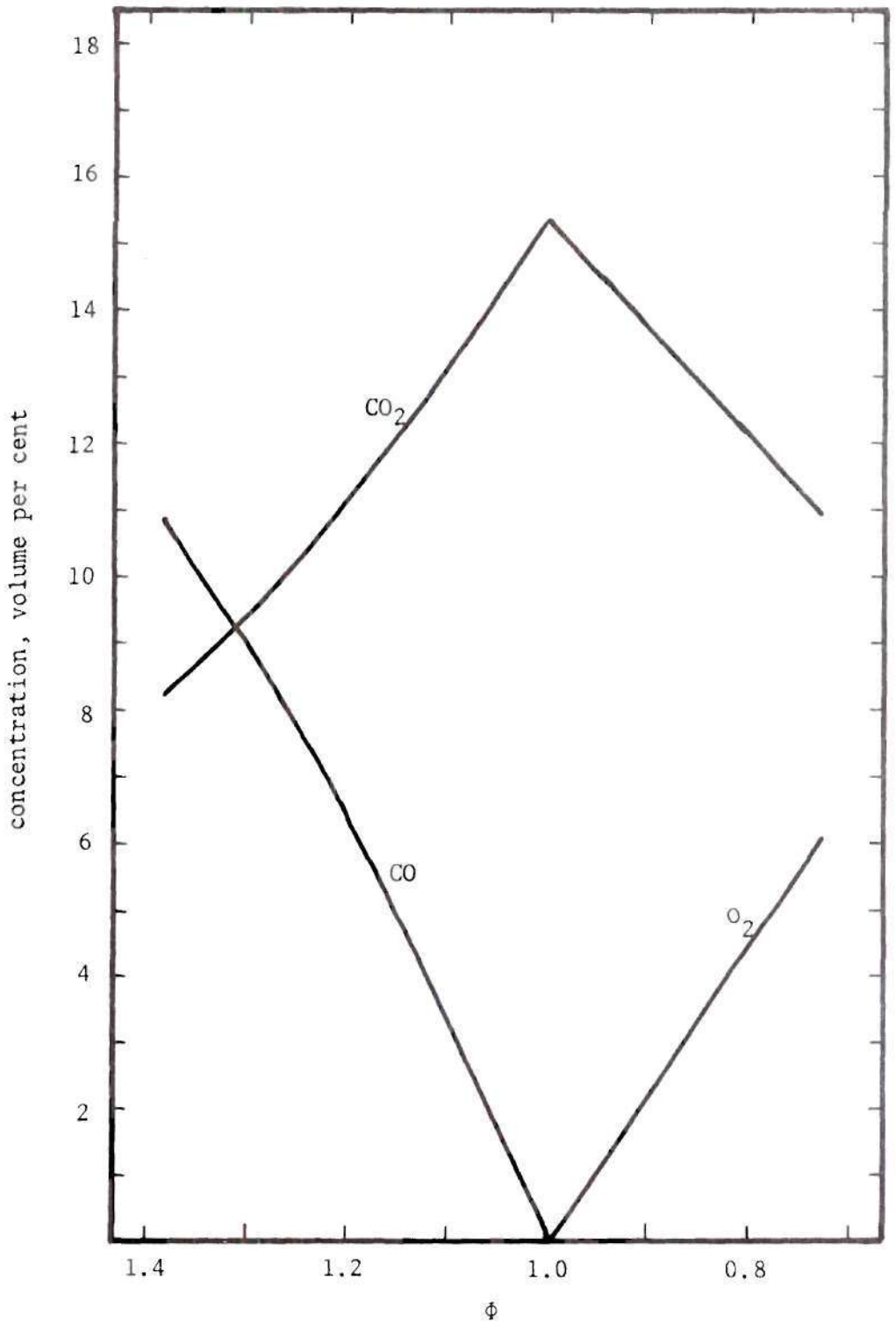

Figure 30. D'Alleva Chart for Indolene, $CH_{1.86}$

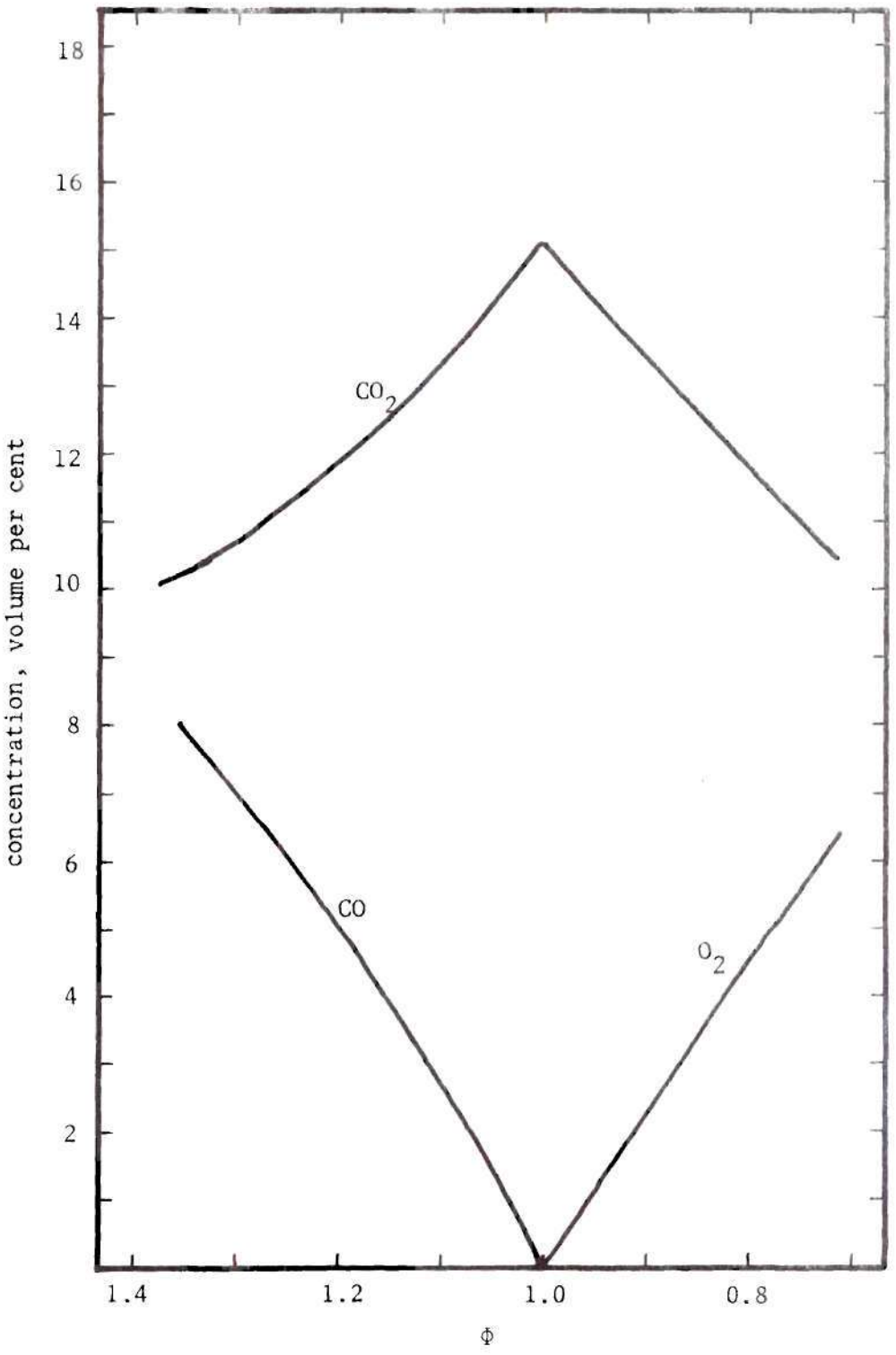

Figure 31. D'Alleva Chart for Methanol, CH_4O

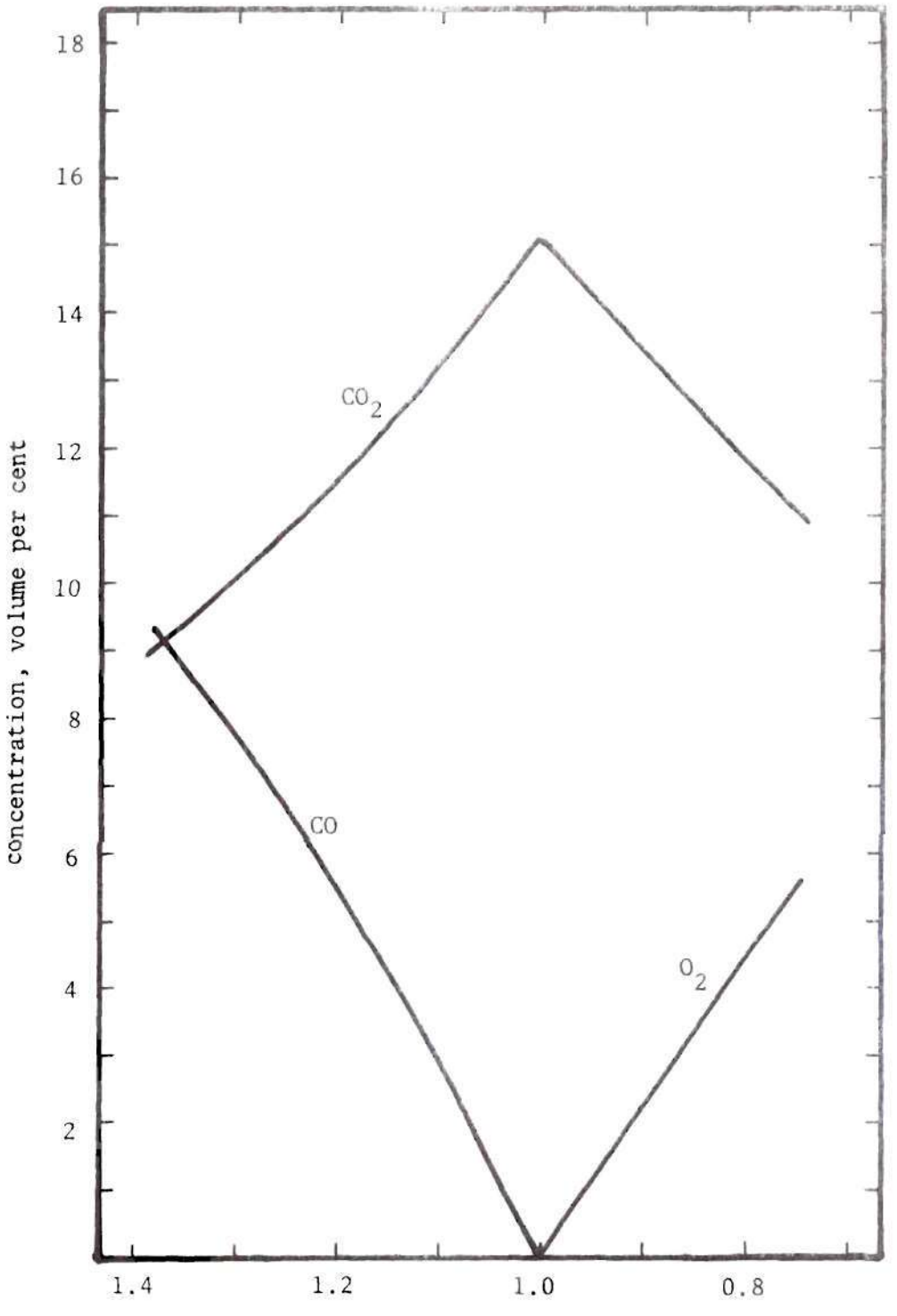

Figure 32. D'Alleva Chart for Ethanol, C_2H_6O

APPENDIX D

DERIVATION OF A METHOD TO CALCULATE AIR-FUEL RATIOS FOR ALCOHOL FUELS USING THE SPINDT METHOD

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DERIVATION OF A METHOD TO CALCULATE AIR-FUEL RATIOS FOR ALCOHOL FUELS USING THE SPINDT METHOD

The D'Alleva charts (39) for calculating air-fuel ratios from exhaust analysis is only one of several methods in use. The primary limitation of the D'Alleva charts is the assumption of complete combustion. R. S. Spindt has derived a method which yields reasonable results even with poor combustion (40). In his work, Spindt derives a formula for the air-fuel ratio in terms of the exhaust constituents for pure hydrocarbon fuels. This formula could be applied to Indolene calculations. However, it became necessary to derive a similar expression for alcohol fuels.

Consider an alcohol fuel molecule to be of the form CH_yO_z. Allow 100 moles of air to react with the fuel. The general combustion equation becomes:

$$(20.99 + \frac{z}{2})O_2 + C + \frac{y}{2}H_2 + 79.01N_2 = aCO_2 + bCO + cH_2O + dH_2 + eO_2 + 79.01\,N_2 \qquad (1)$$

where 1, y, and z represent the atoms of carbon, hydrogen, and oxygen in the fuel that undergoes combustion, and a, b, c, d, and e represent moles of products.

Balancing atoms:

Carbon: $1 = a + b$ (2)

Hydrogen: $y = 2c + 2d$ (3)

Oxygen: $20.99 + \frac{z}{2} = a + \frac{b}{2} + \frac{c}{2} + e$ (4)

The water-gas equilibrium equation may be written:

$$\frac{(c)(b)}{(a)(d)} = K \quad , \tag{5}$$

where K equals 3.5, and is defined as the water-gas equilibrium constant.

The air-fuel ratio may be represented as:

$$AF = \frac{28.97(100)}{12.01 + 1.008y + 16z} \tag{6}$$

The mass fractions of carbon (F_C), hydrogen (F_H), and oxygen (F_O), are defined as:

$$F_c = \frac{12.01}{12.01 + 1.008y + 16z} \tag{7a}$$

$$F_H = \frac{1.008y}{12.01 + 1.008y + 16z} \tag{7b}$$

$$F_o = \frac{16z}{12.01 + 1.008y + 16z} \tag{7c}$$

Solving equations (6) and (7a, 7b, 7c) for 1, y, and z yields:

$$1 = \frac{2897\ (F_C)}{12.01\ AF} = \frac{241.2\ (F_C)}{AF} \tag{8}$$

$$y = \frac{2897\ (F_H)}{1.008\ AF} = \frac{2874\ (F_H)}{AF} \tag{9}$$

$$z = \frac{2897\ (F_O)}{16\ AF} = \frac{181.1\ (F_O)}{AF} \tag{10}$$

The moles of exhaust gas of an alcohol fuel are equal to the moles of the original charge times the fraction unburned plus the moles of products times the fraction burned. Defining B as the mole fraction burned, and U as the mole fraction unburned, we may write:

$$B + U = 1 \tag{11}$$

Defining T as the total moles of exhaust gas per 100 moles of air allows us to write expressions for the relative concentrations of the exhaust constituents.

$$(CO_2) = \frac{a(100B)}{T} \tag{12}$$

$$(CO) = \frac{b(100B)}{T} \tag{13}$$

$$(CH) = \frac{(100U)}{T} \tag{14}$$

where (CH) = concentration of unburned hydrocarbons on a carbon atom basis, and

$$(O_2) = 100\left[\frac{(20.99 + \frac{z}{2})U + E(B)}{T}\right] \tag{15}$$

Define the ratios R and Q to be:

$$R = \frac{CO}{CO_2} = \frac{b}{a} \tag{16}$$

$$Q = \frac{O_2}{CO_2} = \frac{(20.99 + \frac{z}{2})U + E(B)}{a(B)} \tag{17}$$

From equations (2), (12), (13), and (14), we may write:

$$\frac{CH}{CO + CO_2} = \frac{U}{B} \tag{18}$$

From equations (11) and (18):

$$U = \frac{CH}{CO + CO_2 + CH} \tag{19}$$

$$B = \frac{CO + CO_2}{CO + CO_2 + CH} \tag{20}$$

When solved for an expression for E, equation (17) becomes:

$$E = aQ - (20.99 + \frac{z}{2}) \frac{U}{B} \tag{21}$$

We may also write expressions for a, b, and c from equations (2), (3), (5), and (16) as follows:

$$a = \frac{1}{1 + R} \tag{22}$$

$$b = \frac{R}{1 + R} \tag{23}$$

$$c = \frac{K\ y\ /\ 2}{R + K} \tag{24}$$

Substituting equations (21), (22), (23), and (24) into equation (4) yields the following expression:

$$20.99 + \frac{z}{2} = \frac{1}{R+1} + \frac{R}{2(R+1)} + \frac{yK}{4(R+K)} + \frac{Q}{R+1} - \frac{20.99U}{B} - \frac{zU}{2B} \qquad (25)$$

Inserting equations (8), (9), and (10), and (11) into equation (25), we obtain the following expression for the air-fuel ratio in terms of the exhaust gas components.

$$AF = B\left[\frac{11.491\ F_c}{R+1}\left\{1 + \frac{R}{2} + Q - \frac{z(R+1)}{2B}\right\} + \frac{120\ F_H}{R+K}\right] \qquad (26)$$

APPENDIX E

DATA TABLES

Table E1. Engine Performance and Exhaust Mass Data at 1000 rpm with Indolene at MPSA

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
1.274	180.6	83.20	10637	4.34	.68	422.0	257.7
1.252	179.5	84.44	10333	3.87	1.20	432.1	248.1
1.210	190.2	87.95	10089	3.25	2.02	463.5	257.2
1.190	191.3	87.15	10049	3.03	1.12	486.6	216.0
1.190	192.7	88.89	10008	2.94	2.22	482.9	232.7
1.190	191.3	86.40	10191	2.92	2.44	492.1	237.7
1.190	193.5	93.68	9724	2.71	1.71	476.4	260.6
1.152	193.5	94.77	9358	2.57	1.92	493.4	189.5
1.143	180.6	85.45	9399	3.69	1.09	510.4	127.9
1.134	196.6	92.17	9500	2.77	2.08	518.9	185.0
1.126	179.5	85.92	9216	3.29	1.18	525.8	84.99
1.100	190.2	87.98	9135	2.70	2.18	546.9	109.0
1.100	193.5	93.73	8932	2.31	4.86	533.8	108.9
1.072	196.6	92.66	8952	2.44	4.82	574.0	70.76
1.068	180.6	85.37	8831	3.04	2.19	578.4	24.02
1.068	192.4	89.40	8912	2.37	2.40	576.3	70.33
1.060	191.3	86.54	9054	2.11	3.10	601.7	28.73
1.052	179.5	84.82	8709	2.63	1.97	589.7	13.29
1.052	193.5	94.87	8546	2.30	4.67	572.7	56.93
1.045	191.3	89.14	8770	2.25	1.89	604.1	35.56
1.045	193.5	94.65	8506	2.05	7.67	582.5	36.42
1.045	190.2	87.24	8912	1.89	7.18	614.1	6.00
1.023	192.4	88.86	8709	1.54	8.57	628.6	5.42
.974	193.5	93.63	7998	1.79	11.90	609.9	10.35
.943	196.6	89.31	8201	1.63	18.04	621.4	3.28
.937	190.2	88.59	7876	1.58	15.71	596.8	4.25
.937	193.5	90.04	7958	1.33	-- **	602.0	4.02
.896	191.3	82.27	8181	1.43	14.22	616.4	2.88
.896	193.5	86.31	7897	1.35	18.25	593.5	4.18
.880	191.3	85.04	7978	1.42	14.34	602.5	2.86
.849	180.6	77.62	7633	2.11	11.23	577.2	2.85
.849	190.2	80.99	7937	1.46	19.13	597.7	2.95
.839	192.4	79.93	7876	1.16	15.07	590.4	2.96
.830	190.2	79.03	7795	1.47	-- **	587.3	2.97
.830	196.6	83.39	7856	1.42	17.30	590.7	2.98
.820	179.5	76.41	7592	2.03	8.91	571.1	2.92
.820	191.3	73.91	8161	1.29	12.39	612.3	3.14
.807	191.3	76.02	8059	1.37	16.56	604.0	3.15
.785	193.5	78.47	7633	1.17	12.58	572.8	4.58
.785	192.4	76.10	7694	1.08	10.65	577.5	3.08
.764	179.5	71.34	7369	1.81	12.53	556.0	3.05
.764	190.2	71.48	7816	1.45	11.39	587.9	3.23
.748	190.2	65.22	8425	1.38	11.26	599.9	3.56

Table E1. Continued

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
.741	108.6	71.29	7430	2.04	13.00	557.5	3.16
.741	190.2	71.65	7816	1.57	15.87	586.0	3.33
.691	196.6	73.49	7288	1.40	7.36	548.0	3.30

*indicated specific energy consumption
**Exhaust emissions from engine exceeded maximum meter reading

Table E2. Engine Performance and Exhaust Mass Data at 1800 rpm with Indolene at MPSA

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
1.263	212.8	54.95	10130	4.11	.86	418.2	242.3
1.252	215.0	64.41	9237	3.60	.80	388.9	196.4
1.210	228.9	60.50	9785	3.10	1.54	446.6	250.3
1.210	227.7	65.07	9277	2.87	1.21	422.2	246.4
1.134	227.7	62.39	8891	2.41	1.77	287.0	162.4
1.126	228.9	64.04	8810	2.60	2.30	495.0	145.0
1.100	226.2	58.61	9013	3.61	1.79	536.0	111.9
1.076	227.7	61.96	8485	2.24	4.02	538.2	75.47
1.076	227.7	61.61	8506	1.87	7.78	538.3	75.62
1.052	227.7	64.51	8100	1.82	3.43	543.7	35.55
1.045	226.2	57.38	8749	2.38	5.05	601.0	29.43
1.030	226.2	60.00	8425	2.06	6.63	601.7	13.02
.949	226.2	58.51	7876	1.30	10.75	600.4	5.74
.908	226.2	53.87	8039	1.38	12.12	606.2	2.80
.859	226.2	57.56	7511	1.08	12.22	567.2	2.76
.830	226.2	51.10	7856	.97	6.37	590.2	2.98
.820	226.2	56.69	7430	1.32	10.24	556.8	2.86
.807	226.2	51.74	7714	1.23	3.97	578.3	3.02

*indicated specific energy consumption

Table E3. Engine Performance and Exhaust Mass Data at 1000 rpm with Methanol at MPSA

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
1.095	158.9	82.04	8813	1.52	5.61	449.2	36.3
1.095	161.9	84.86	8688	1.27	6.86	440.0	44.95
1.082	166.1	85.89	8813	1.72	6.47	448.2	72.27
1.059	168.9	87.64	8582	1.22	5.63	484.0	60.2
1.052	161.9	84.57	8458	1.19	7.43	466.2	12.64
1.044	166.1	85.42	8515	1.63	7.41	471.4	44.03
1.027	166.1	83.59	8534	1.44	10.58	492.4	19.31
1.022	168.9	85.51	8573	1.06	13.90	536.1	20.9
.967	166.1	83.58	8112	.96	-- **	493.7	6.50
.954	158.9	83.30	7603	.91	-- **	462.6	9.27
.933	168.9	82.34	8074	.92	14.70	493.0	8.97
.924	158.9	79.22	7632	.82	-- **	466.1	4.31
.904	168.9	78.60	8160	.78	18.90	498.1	7.0
.898	166.1	78.64	7862	.83	13.86	480.4	6.84
.898	161.9	79.85	7622	.76	14.10	452.6	3.48
.859	161.9	77.05	7507	.77	13.70	460.9	3.43
.847	176.3	74.78	7786	.77	11.07	476.6	7.23
.831	168.9	74.34	7920	.76	10.90	488.0	7.50
.831	161.9	75109	7344	.76	14.10	452.6	3.48
.820	161.9	73.15	7459	.73	12.60	458.8	3.57
.781	161.9	71.48	7258	.75	11.61	448.4	3.70
.771	161.9	71.21	7219	.72	8.89	444.7	3.72
.768	176.3	73.09	7872	.78	8.37	485.7	8.14
.762	168.9	68.41	7786	.66	9.35	477.0	8.14
.739	176.3	67.18	7949	.76	3.66	490.5	8.59
.728	161.9	68.33	7162	.78	8.37	441.5	3.92

*indicated specific energy consumption
**Exhaust emissions from engine exceeded maximum meter reading

Table E4. Engine Performance and Exhaust Mass Data at 1800 rpm with Methanol at MPSA

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
1.182	196.6	61.50	8842	1.75	1.09	369.0	134.0
1.160	195.4	60.25	8698	.87	1.41	372.4	139.6
1.150	198.5	61.05	8650	.72	2.46	391.3	116.6
1.140	202.7	59.87	8880	1.24	2.12	396.0	131.8
1.126	199.6	58.93	8717	1.51	2.29	403.0	102.6
1.124	197.4	60.97	8390	2.47	2.35	386.5	118.2
1.124	199.6	60.58	8563	1.55	1.29	397.2	100.9
1.120	204.5	63.08	8506	1.34	1.76	399.4	93.7
1.092	199.6	60.23	8333	1.04	2.10	413.9	74.6
1.073	198.5	61.25	8045	.63	2.22	418.3	54.0
1.073	197.4	61.39	7997	1.93	2.73	414.9	63.0
1.073	199.6	59.33	8294	1.28	2.96	433.1	52.0
1.059	204.5	60.35	8237	1.29	3.80	442.4	40.2
1.056	198.5	60.25	8006	.77	1.45	429.5	43.0
1.051	199.6	60.58	8035	1.32	1.78	438.1	35.6
1.029	197.4	61.55	7709	1.57	3.35	445.4	13.59
1.027	199.6	60.23	7872	.82	3.75	455.1	11.89
1.025	198.5	59.36	7939	.67	4.28	461.0	8.02
.997	198.5	56.61	7882	.62	5.20	478.2	8.10
.994	199.6	59.98	7661	.70	7.33	467.8	4.37
.976	199.6	58.29	7680	1.05	4.51	466.5	4.47
.971	204.5	58.83	7786	1.10	4.10	474.9	4.55
.954	198.5	56.24	7661	.41	6.67	467.6	2.09
.933	198.5	57.65	7478	.57	6.85	458.7	2.08
.920	199.6	56.03	7565	.98	6.86	464.4	2.13
.920	197.4	56.77	7411	.96	6.89	454.6	2.51
.920	199.6	57.01	7411	.51	8.26	454.3	2.10
.897	202.7	54.35	7690	.48	6.17	471.1	2.24
.880	199.0	56.21	7344	.88	6.46	450.1	2.18
.880	201.6	54.51	7546	.48	5.79	462.0	2.23
.880	201.6	55.15	7536	.37	7.07	461.0	2.23
.879	197.4	55.28	7382	.77	5.96	452.4	2.64
.867	204.5	52.49	7709	1.12	8.28	473.1	2.34

*indicated specific energy consumption

Table E5. Engine Performance and Exhaust Mass Data at 1000 rpm with Ethanol at MPSA

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
1.179	177.1	88.00	9761	4.31	.57	406.8	197.4
1.134	176.1	86.18	9326	4.25	1.16	421.8	172.1
1.103	175.5	87.26	8967	3.09	5.03	451.4	67.9
1.103	169.3	82.79	8826	2.97	3.19	445.1	64.7
1.099	170.4	81.90	9005	3.47	4.00	456.8	59.8
1.093	177.0	84.57	9069	2.79	4.59	466.8	67.1
1.093	180.6	82.83	9134	2.94	4.45	472.0	58.8
1.085	179.6	85.24	9095	3.31	3.72	476.2	58.9
1.073	184.3	85.24	9223	4.14	1.26	495.1	53.3
1.073	177.1	87.12	8941	3.42	2.07	477.0	69.2
1.067	175.5	86.59	8800	3.75	1.27	479.3	50.8
1.064	191.5	87.17	9274	4.30	6.06	476.2	33.9
1.042	175.5	84.69	8647	2.36	6.07	499.0	20.10
1.042	171.9	84.35	8596	1.97	4.61	465.5	4.80
1.039	179.6	84.23	8890	2.06	5.51	520.5	9.26
1.039	177.0	84.06	8711	2.06	5.77	508.6	14.00
1.038	180.6	83.87	8775	2.28	5.96	512.3	18.23
1.036	191.5	84.57	9300	3.91	8.73	546.8	7.30
1.030	175.5	81.14	8711	2.47	4.73	520.9	6.90
1.023	184.3	68.49	8813	2.95	5.34	533.0	6.97
1.018	177.1	83.97	8634	2.24	10.89	526.7	6.85
1.018	175.5	84.23	8416	.98	4.21	514.4	4.89
.928	177.0	80.38	8147	1.52	14.20	544.0	2.74
.928	175.5	80.44	8147	1.31	10.06	486.6	2.66
.924	180.6	80.76	8173	1.28	11.98	519.0	2.40
.901	183.7	81.06	8250	1.91	11.84	523.4	2.98
.901	171.9	79.24	7865	1.53	11.49	499.4	2.85
.901	179.6	77.90	8288	1.38	11.28	525.9	2.49
.896	177.0	78.07	8147	1.45	12.37	519.5	2.46
.866	175.5	76.21	8070	1.49	12.10	514.6	2.54
.845	191.5	77.73	8455	2.69	13.26	541.5	2.73
.845	178.6	74.93	8019	1.52	14.20	544.0	2.74
.822	177.8	74.45	7827	1.47	9.21	499.9	2.65
.818	179.6	73.64	8045	1.40	11.09	513.7	2.71
.818	180.6	72.09	7981	1.31	12.68	481.2	2.53
.800	177.0	50.51	7968	1.51	7.86	511.0	2.74
.796	189.8	78.65	8122	1.80	12.21	519.9	2.80
.786	178.6	70.30	7942	1.34	8.47	507.7	2.78
.759	189.8	70.08	8224	1.87	8.12	527.6	3.01
.759	179.6	67.33	7955	1.51	7.55	505.5	2.87
.759	178.6	70.43	7840	1.37	8.91	502.5	2.87
.747	177.0	67.86	7904	1.65	5.68	506.7	2.94
.740	190.8	68.86	8340	2.79	11.04	535.7	3.12
.726	178.6	65.98	7878	1.53	4.95	506.1	3.01

Table E5. Continued

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
.726	184.1	65.90	8147	1.59	4.61	523.6	3.11
.711	177.8	67.13	7519	1.76	6.86	485.0	2.94
.711	186.8	66.62	7993	1.66	8.21	516.0	3.11
.689	189.8	62.87	8275	2.27	4.67	532.0	3.35
.689	186.8	59.63	8275	1.74	3.85	532.3	3.36

*indicated specific energy consumption

Table E6. Engine Performance and Exhaust Mass Data at 1800 rpm with Ethanol at MPSA

Equil Ratio	Airf g/min	imep psi	isec* btu/ihp-h	HC g/ihp-h	NO_x g/ihp-h	CO_2 g/ihp-h	CO g/ihp-h
1.293	196.2	56.19	9787	6.23	.54	325.6	205.2
1.244	200.9	58.17	9351	5.46	.74	343.0	165.1
1.211	199.1	64.48	8596	4.78	.50	336.7	149.3
1.156	199.1	62.86	8288	4.38	.81	372.9	110.9
1.149	200.9	60.52	8352	4.87	.74	373.8	121.0
1.127	217.9	57.06	9287	3.66	1.48	445.8	59.7
1.127	213.6	55.75	9403	2.85	2.39	444.1	96.6
1.123	219.7	60.47	9082	2.73	2.69	434.9	87.3
1.113	220.4	59.68	9057	2.62	2.88	445.0	76.4
1.113	215.5	61.64	8672	2.58	2.75	424.7	77.5
1.099	200.9	60.63	7981	3.48	1.06	401.2	74.1
1.093	219.7	60.30	8813	2.56	3.21	453.7	63.1
1.085	198.5	62.75	7750	3.42	1.26	404.1	61.3
1.085	217.9	57.26	8954	2.90	3.88	474.1	23.6
1.085	214.3	59.82	8583	2.28	3.66	447.3	61.6
1.076	219.2	62.95	8378	2.67	4.84	449.8	41.1
1.076	215.5	61.58	8672	2.58	2.75	424.7	77.5
1.076	213.6	56.12	8826	2.44	2.90	471.8	49.8
1.065	200.9	64.83	7507	3.22	2.12	411.8	35.98
1.064	217.9	54.84	9146	1.30	4.44	508.9	2.35
1.048	211.2	59.95	8173	1.99	4.42	466.1	23.1
1.039	219.7	60.20	8378	2.08	4.44	485.2	32.9
1.038	219.2	62.64	8122	1.76	5.83	443.1	7.88
1.018	213.6	56.66	8339	1.77	4.51	507.6	13.2
1.018	215.5	61.58	7929	1.46	5.16	483.2	7.95
.979	211.2	60.46	7673	1.28	6.58	486.4	3.16
.953	219.7	55.43	8211	1.32	7.81	521.0	2.79
.953	213.6	55.81	7955	1.12	7.09	504.9	2.24
.943	215.5	56.75	7840	1.04	7.74	497.3	2.25
.928	219.0	55.91	7968	1.16	7.62	506.9	2.32
.924	219.2	54.23	8019	1.34	8.68	509.9	2.35
.896	213.6	57.37	7532	1.02	7.49	480.6	2.27
.896	211.2	56.26	7519	1.01	8.22	479.1	2.27
.878	218.5	56.93	7507	1.16	5.92	478.8	2.34
.866	215.5	50.56	8032	1.03	7.68	512.3	2.53
.862	213.6	50.34	7929	.98	5.35	504.7	2.51
.853	219.0	49.92	8096	1.11	5.53	517.4	2.59
.840	217.9	56.35	7315	1.21	5.87	567.5	2.37
.833	224.7	50.76	8057	1.13	4.13	515.4	2.64
.818	213.6	51.37	7545	1.08	5.40	481.2	2.53
.818	216.7	51.88	7673	1.08	4.01	489.3	2.58
.804	215.5	51.97	7494	1.07	4.00	478.6	2.56
.783	219.0	51.93	7468	1.17	4.32	479.1	2.63
.783	224.7	52.03	7622	1.17	2.87	489.1	2.68

*indicated specific energy consumption

APPENDIX F

SAMPLE CALCULATION OF MASS EMISSIONS FROM DRY VOLUME CONCENTRATIONS

APPENDIX F

SAMPLE CALCULATION OF MASS EMISSIONS FROM DRY VOLUME CONCENTRATIONS

Fuel = Methanol = CH_4O

ihp = 3.85

AF = 6.90

$\dot{w}_f$ = 24.48 g/min

(HC) = dry volume per cent HC on a carbon atom basis = .036%

(NO_x) = dry volume per cent NO_x = .40%

(CO_2) = dry volume per cent CO_2 = 14.0%

$(HC)_w$ = wet volume per cent HC on a carbon atom basis

$(NO_x)_w$ = wet volume per cent NO_x

$(CO_2)_w$ = wet volume per cent CO_2

$(CO)_w$ = wet volume per cent CO

Step 1:

The correction factor, C_f, used in reference (41) to convert dry concentrations to a wet basis is given as:

$$C_f = \frac{(HC)_w}{(HC)}$$

where

$$(HC)_w = \frac{(HC)}{a - (HC)b}$$

$$a = 1 + \frac{\frac{y}{2n}}{\frac{CO}{3.8CO_2} + 1} = 1 + \frac{\frac{2}{N}}{\frac{CO}{3.8CO_2} + 1}$$

$$b = \frac{.07}{\frac{CO}{3.8CO_2} + 1}$$

$$N = \frac{100}{(HC) + (CO) + (CO_2)}$$

Substituting in appropriate dry volume concentrations yields:

$N = 6.925$

$a = 1.287$

$b = 0.0695$

$(HC)_w = 0.280$

$C_f = .7778$

Step 2:

Multiply all dry volume concentrations by the correction factor, C_f, to obtain the wet volume concentrations for each emission.

$(HC)_w = (.14)(0.8821) = 0.0280$

$(NO_x)_w = (.28)(0.8821) = 0.3115$

$(CO_2)_w = (15.0)(0.8821) = 10.91$

$(CO)_w = (.40)(0.8821) = 0.312$

Step 3:

Mass emissions on a $\left(\frac{gm}{iph - h}\right)$ basis are now obtained.

$$\frac{\text{gm emission}}{\text{ihp - h}} = \frac{(\dot{w}_f \frac{\text{gm}}{\text{min}})(60 \frac{\text{min}}{\text{hr}})(1 + AF)(\text{emission})_w M_e}{(\text{ihp})(M_{mix})}$$

where M_e = molecular weight of particular emission $\left(\frac{\text{lb}_m}{\text{lb}_{mole}}\right)$

M_{mix} = molecular weight of AF mixture

$= \frac{928(1 + AF)}{32AF + 29}$ for methanol.

Solving for the appropriate mass emissions:

$$\frac{\text{gm HC}}{\text{ihp-h}} = 0.92$$

$$\frac{\text{gm NO}_x}{\text{ihp-h}} = 14.7$$

$$\frac{\text{gm CO}_2}{\text{ihp-h}} = 493.0$$

$$\frac{\text{gm CO}}{\text{ihp-h}} = 8.97$$

It should be noted that this method of calculation of mass emissions from dry volume concentrations is equally applicable to alcohol fuels. Stivender's derivation includes an expression for the concentration of water which follows only from the concentration of products of hydrogen in the combustion reaction and the water-gas equilibrium equation. The expression for the concentration of water determines the correction factor to convert from a dry to wet mass basis. Therefore, the extra oxygen atom(s) found in the alcohol fuel

molecule do not alter his derivation, but are accounted for in the relative concentrations of products in the combustion reaction equation and in the water-gas equilibrium equation.

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